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Vice President.....W. J. Franklin
Secretary.....J. L. Brown
Treasurer.....J. W. Thompson
Board of Governors: R. H. Combs, Luther Granderson, Jr., H. H. Himstedt, J. C. Lewis, C. H. Miller

Atlanta

Organized 1937
Headquarters, Atlanta, Ga.

President.....J. M. Lazenby
Vice President.....T. A. Barrow, Jr.
Secretary.....F. W. Bull
Treasurer.....J. A. Marshall
Board of Governors: J. G. Croley, W. J. McKinney

Baltimore

Organized 1949
Headquarters, Baltimore, Md.

President.....E. H. Taze
Vice President.....E. J. Morris
Secretary.....R. F. Weisman
Treasurer.....W. W. Hill
Board of Governors: G. M. Baldwin, H. D. Glaser, A. J. Simpson

Bluegrass

Organized 1954
Headquarters, Louisville, Ky.

President.....J. W. May
Vice President.....W. M. Pace
Secretary.....A. E. Hausmann
Treasurer.....R. B. Duggan
Board of Governors: H. L. Carr, W. J. Killian, H. A. Zanone

British Columbia

Organized 1952
Headquarters, Vancouver, B. C., Canada

President.....S. C. Gale
Vice President.....Cornelius Van Boeyen
Secretary.....D. A. English
Treasurer.....R. D. Hale
Board of Governors: J. M. Bird, D. A. English, E. H. Norton, J. C. Thompson, D. W. Thomson

Central New York

Organized 1944
Headquarters, Syracuse

President.....F. D. Putnam
Vice President.....L. A. Childs
Secretary.....Merle Weninger
Treasurer.....H. J. Morton
Board of Governors: H. K. Ormsby, Jr., K. T. Sprague

Central Ohio

Organized 1944
Headquarters, Columbus

President.....J. A. Guy
Vice President.....R. A. Wilson
Secretary.....W. A. Schoonover
Treasurer.....T. R. Walker
Board of Governors: J. W. Ford and E. T. Stluka

Cincinnati

Organized 1932
Headquarters, Cincinnati, Ohio

President.....F. W. Wilson
Vice President.....A. H. Gerdsen
Secretary.....R. G. Anderson
Treasurer.....T. D. Reiley
Board of Governors: C. P. Krantz

Connecticut

Organized 1940
Headquarters, New Haven

President.....C. L. L'Hommedieu
Vice President.....E. J. Hoagland
Secretary.....Walter Heywood
Treasurer.....Carl Wahnquist, Jr.
Board of Governors: Fritz Honerkamp, G. F. Nieske, J. D. Pierce

Delta

Organized 1939
Headquarters, New Orleans, La.

President.....C. V. Blankston
Vice President.....H. N. Stall
Secretary.....W. B. Martin, Jr.
Treasurer.....T. J. Jenny
Board of Governors: J. T. Knight, Jr. and I. A. Lindberg

Empire State Capital

Organized 1951
Headquarters, Albany, N. Y.

President.....E. C. Doyle
1st Vice President.....G. G. Davis
2nd Vice President.....L. V. Appleby
Secretary-Treasurer.....H. F. Kruger
Board of Governors: E. J. Mahoney, R. F. Peck, F. W. Sheldon

Local Chapter Officers—1954 *(Continued)*

Golden Gate

Organized 1937
Headquarters, San Francisco, Calif.

President.....Eric Roberts
Vice President.....D. E. McLeod
Secretary.....F. K. Crouch
Treasurer.....Herb Duncan
Board of Governors: James McLachlan, J. E. Murray, J. B. Smith

Illinois

Organized 1906
Headquarters, Chicago

President.....G. V. Zintel
Vice President.....G. G. Freyder
Secretary.....H. G. Gragg
Treasurer.....J. F. Cummiskey
Board of Governors: H. E. Anderson, E. P. Heckel, Jr., H. C. Stevens

Indiana

Organized 1943
Headquarters, Indianapolis

President.....P. O. Patterson
Vice President.....J. W. Jackson
Secretary.....A. O. Roche, Jr.
Treasurer.....F. B. Morse
Board of Governors: R. C. Blackman, W. W. Grear, G. W. Vogel

Inland Empire

Organized 1950
Headquarters, Spokane, Wash.

President.....H. A. Bickel
Vice President.....F. W. Jenkinson
Secretary.....Max Tonn
Treasurer.....J. A. Doyle
Board of Governors: D. E. Arnold and K. M. Wood

Iowa

Organized 1940
Headquarters, Des Moines

President.....W. E. Nanes
Vice President.....G. J. Kraai
Secretary-Treasurer.....D. E. Schroeder
Board of Governors: H. E. Drain, P. R. Duitch, J. F. Sandfort

Kansas

Organized 1951
Headquarters, Wichita

President.....Charles Yoe
Vice President.....A. B. Newton
Secretary.....T. L. Roberts
Treasurer.....H. M. Skalla
Board of Governors: R. F. Baurer, O. P. Bullock, R. L. Pennington

Kansas City

Organized 1917
Headquarters, Kansas City, Mo.

President.....F. K. Ladewig
Vice President.....A. S. Hurt, Jr.
Secretary.....R. M. Spencer
Treasurer.....S. C. McCann
Board of Governors: C. S. Graves, Henry Nottberg, Jr., G. H. Stoffer

Manitoba

Organized 1935
Headquarters, Winnipeg, Man., Canada

President.....G. T. Christie
Vice President.....G. C. Davis
Secretary-Treasurer.....A. J. McIntyre
Board of Governors: J. A. Bell, R. G. Cawker, P. L. Charles, R. M. Fraser, A. K. Piercy, H. F. Randall, J. C. Stangl, D. S. Swain

Massachusetts

Organized 1912
Headquarters, Boston

President.....G. W. Sprague
Vice President.....G. D. Fife
Secretary.....W. G. Burbo
Treasurer.....W. G. Martin, Jr.
Board of Governors: J. A. Ivester, E. M. Tracey, R. G. Vanderweil

Memphis

Organized 1944
Headquarters, Memphis, Tenn.

President.....H. H. Wilson
Vice President.....John Hilton, II
Secretary.....A. T. Bevil
Treasurer.....E. A. Smith
Board of Governors: A. T. Bevil, W. L. Henson, John Hilton, II, A. W. Shelby, E. A. Smith, W. E. Thorpe, H. H. Wilson

Miami Valley

Organized 1950
Headquarters, Dayton, Ohio

President.....R. W. Kimmel
Vice President.....D. E. Tullis
Secretary.....J. B. Rishel
Treasurer.....L. P. Brehm, Jr.
Board of Governors: R. C. Allen, C. T. Doudican, W. W. Hackney, J. M. Schweiger

Michigan

Organized 1916
Headquarters, Detroit

President.....E. F. Glanz
Vice President.....D. S. Falk
Secretary.....J. H. Spitzley
Treasurer.....J. N. Livermore
Board of Governors: P. S. Hosman, R. H. Oberschulte, Axel Marin, G. S. Whittaker

Local Chapter Officers—1954 *(Continued)*

Minnesota

Organized 1918
Headquarters, Minneapolis

President.....J. A. Craig
Vice President.....J. S. Locke
Secretary.....E. T. Erickson
Treasurer.....F. G. Vogt
Board of Governors: G. M. Kendrick, William Sturm

Mississippi

Organized 1953
Headquarters, Jackson

President.....I. E. Rowe
Vice President.....J. E. Davis, Jr.
Secretary.....M. E. Price
Treasurer.....T. E. Stepan
Board of Governors: Burt Lomax, Jr., F. H. North, O. F. Rogers

Montreal

Organized 1936
Headquarters, Montreal, Que., Canada

President.....D. L. Lindsay
Vice President.....R. J. Ker
Secretary.....S. R. Plamondon
Treasurer.....W. E. Jarvis
Board of Governors: Robert Clapperton, B. J. Horsburgh, R. E. J. Layton, G. N. Martin, H. G. S. Murray

Nebraska

Organized 1940
Headquarters, Omaha

President.....C. A. Goth
Vice President.....T. E. Davis
Secretary.....L. J. Paulsen
Treasurer.....F. E. Hawley, R. D. Anderson*
Board of Governors: S. W. Black, O. J. Smith

New Mexico

Organized 1954
Headquarters, Albuquerque

President.....J. H. Van Alsburg
Vice President.....H. F. Munn
Secretary.....J. K. James
Treasurer.....R. W. Haines
Board of Governors: F. H. Bridgers, C. R. Wherritt

New York

Organized 1911
Headquarters, New York

President.....R. L. Stinard
Vice President.....Albert Giannini
Secretary.....Carl H. Flink
Treasurer.....W. M. Heebner
Board of Governors: H. S. Birkett, J. B. Hewett, C. R. Hiers, W. O. Huebner

North Jersey

Organized 1952
Headquarters, Newark, N. J.

President.....H. M. Patrick
Vice President.....C. H. Smith
Secretary.....W. C. Kruse, Jr.
Treasurer.....M. C. Christesen
Board of Governors: F. H. Faust, H. P. Morehouse, S. H. Nitzberg, G. C. Norman

North Texas

Organized 1938
Headquarters, Dallas

President.....M. W. Brown
Vice President.....J. A. Ray
Secretary.....H. G. Gregerson
Treasurer.....G. H. Meffert
Board of Governors: W. E. Frost, E. T. Keck, Jr., P. N. Vinther

Northeastern Oklahoma

Organized 1948
Headquarters, Tulsa

President.....R. W. Winget
Vice President.....R. E. Pauling
Secretary.....J. T. McKinney
Treasurer.....J. D. Ryan
Board of Governors: Fred Colbert, J. C. Netherton, Jr., J. N. Watt

Northern Ohio

Organized 1916
Headquarters, Cleveland

President.....D. E. Mannen
Vice President.....L. C. Burkes
Secretary.....H. R. Canoyer
Treasurer.....R. E. Leising
Board of Governors: J. M. Black, Sherwood Nassau, W. F. D. Neiheiser

Northern Piedmont

Organized 1952
Headquarters, Greensboro, N. C.

President.....R. B. Crosland, Jr.
Vice President.....S. T. Oliver
Secretary.....W. H. Cooper
Treasurer.....R. F. Bean
Board of Governors: F. T. Daddario, L. R. Gorrell, J. E. Hart

Oklahoma

Organized 1935
Headquarters, Oklahoma City

President.....G. E. Ervin
Vice President.....J. R. Patten
Secretary.....F. R. Denham
Treasurer.....A. C. Shelley
Board of Governors: J. H. Carnahan, W. J. Collins, Jr., F. X. Loeffler, Jr.

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* Filled unexpired term.

Local Chapter Officers—1954 *(Continued)*

Ontario

Organized 1922
Headquarters, Toronto, Ont., Canada

President.....M. C. Bailey
Vice President.....D. L. Angus
Secretary-Treasurer.....H. R. Roth
Board of Governors: C. J. Bootes, N. W. Kingsland, Jack Thompson, Charles Torry

Oregon

Organized 1939
Headquarters, Portland

President.....E. E. Kelly
Vice President.....Dick Blankenship
Secretary.....K. N. Flocke
Treasurer.....W. B. Hayes
Board of Governors: R. O. Haneberg, A. N. Hoss, G. G. Van Alst

Ottawa Valley

Organized 1952
Headquarters, Ottawa, Ont., Canada

President.....W. J. Robinson
Vice President.....E. J. Schoenherr
Secretary.....N. J. Howes
Treasurer.....D. W. Banton
Board of Governors: S. J. Bullis, L. J. Greenough, G. L. Ostiguy, J. D. Partington

Pacific Northwest

Organized 1928
Headquarters, Seattle, Wash.

President.....T. C. Caskey
1st Vice President.....I. L. Winsor
2nd Vice President.....L. F. Christofferson
Secretary.....E. W. Triol
Treasurer.....M. W. McKinstry
Board of Governors: J. A. MacDonald, C. L. Timpe

Philadelphia

Organized 1916
Headquarters, Philadelphia, Pa.

President.....L. M. Church
1st Vice President.....A. M. Robertson
2nd Vice President.....C. J. Lubking
Secretary.....C. J. Forve
Treasurer.....P. H. Yeomans
Board of Governors: C. F. Dietz, R. J. Sigel

Pittsburgh

Organized 1919
Headquarters, Pittsburgh, Pa.

President.....E. C. Hach
Vice President.....C. L. Benn
Secretary.....E. H. Riesmeyer, Jr.
Treasurer.....G. W. Cost
Board of Governors: J. H. Allison, A. W. Marshall, C. H. Schneider

Rocky Mountain

Organized 1944
Headquarters, Denver, Colo.

President.....V. F. Vallero
Vice President.....F. C. Allen
Secretary.....D. D. Pearsall
Treasurer.....A. W. Cooper
Board of Governors: P. C. Von Rosenberg, A. S. Widdowfield, H. L. Wray

Sacramento Valley

Organized 1952
Headquarters, Sacramento, Calif.

President.....J. E. Marshall
Vice President.....R. A. Sarro
Secretary.....V. W. Thornburg
Treasurer.....M. J. Delavan
Board of Governors: N. W. Downes, E. C. McKinsey, L. A. O'Meara

St. Louis

Organized 1918
Headquarters, St. Louis, Mo.

President.....H. C. Sharp
1st Vice President.....G. H. Bemarkt
2nd Vice President.....E. W. Clucas
Secretary.....W. P. Norris
Treasurer.....F. E. Ince
Board of Governors: C. H. Burnap, D. E. Hanlon, N. J. Hubbuch, Jr., J. B. Killebrew

Shreveport

Organized 1948
Headquarters, Shreveport, La.

President.....J. S. Malahy, Jr.
Vice President.....R. S. Segall
Secretary.....H. E. Scott
Treasurer.....A. L. Jones, Jr.
Board of Governors: O. J. Dykes, Jr., M. A. Hudson, L. E. Kneipp

South Carolina

Organized 1954
Headquarters, Columbia, S. C.

President.....R. F. Donovan
Vice President.....L. L. Hyatt
Secretary.....J. E. McMurray
Treasurer.....M. W. Rickenbaker, Jr.
Board of Governors: M. R. Durlach, Jr., H. L. McDowell, R. K. Rouse

South Texas

Organized 1938
Headquarters, Houston

President.....J. W. Holland
Vice President.....F. M. Neil
Secretary.....J. C. Lewis
Treasurer.....O. G. Rivoire
Board of Governors: R. S. Sandifer, A. B. Ullrich, Jr.

Local Chapter Officers—1954 *(Continued)*

Southern California

Organized 1930
Headquarters, Los Angeles

President.....C. D. Walz
Vice President.....J. L. McCullough
Secretary.....H. F. Ulovec
Treasurer.....W. J. Biggar
Board of Governors: K. A. Bowman, Robson English, J. R. Hall, A. Z. Levine

Southern Piedmont

Organized 1952
Headquarters, Charlotte, N. C.

President.....R. S. Fullerton
Vice President.....L. F. Lawrence, Jr.
Secretary.....J. R. Clark
Treasurer.....C. M. Setzer, Jr.
Board of Governors: E. S. DeWitt, R. E. Mason, D. P. Schiwetz

Southwest Texas

Organized 1946
Headquarters, San Antonio

President.....L. H. Hornor, Jr.
Vice President.....F. B. Frazee
Secretary.....Boone Crisp
Treasurer.....J. L. Rea
Board of Governors: R. W. Kotzebue, D. E. Locher, C. H. McLeod

Toledo

Organized 1954
Headquarters, Toledo, Ohio

President.....F. C. Richardson, Jr.
Vice President.....G. H. Frost
Secretary.....J. F. Guest
Treasurer.....R. O. Benington, Jr.
Board of Governors: F. A. Edgington, C. F. Hoffman, L. B. LaFrance, D. W. MacDermid

Utah

Organized 1944
Headquarters, Salt Lake City

President.....A. A. Maycock
Vice President.....R. C. Evans
Secretary-Treasurer.....W. L. Stuewe
Board of Governors: A. R. Curtis, M. L. Gollaher, E. J. Watts, D. R. Wilde

Virginia

Organized 1946
Headquarters, Norfolk

President.....H. E. Marquart
Vice President.....T. B. Carpenter
Secretary.....R. L. Burton, Jr.
Treasurer.....C. G. Conaway, Jr.
Board of Governors: M. H. Pelosi, Jr., E. R. Simpson, Jr.

Washington, D. C.

Organized 1935
Headquarters, Washington, D. C.

President.....G. C. F. Asker
Vice President.....W. C. Reamy, Jr.
Secretary.....C. P. Humke
Treasurer.....J. D. Crabtree
Board of Governors: L. T. Davis, J. G. Muirheid, E. F. O'Connell

West Texas

Organized 1953
Headquarters, Lubbock

President.....L. C. McKay
Vice President.....R. B. Carow
Secretary.....J. C. Wharton
Treasurer.....J. G. Samson
Board of Governors: D. G. Halley, O. P. Harlan, Jr.

Western Michigan

Organized 1931
Headquarters, Grand Rapids

President.....F. W. Brundage
Vice President.....S. M. Paganelli
Secretary.....F. L. Murray
Treasurer.....N. A. Buckley
Board of Governors: J. T. Anderson, W. W. Edwards, F. K. Platt

Western New York

Organized 1919
Headquarters, Buffalo

President.....Q. P. Thompson
1st Vice President.....C. W. Kaupp
2nd Vice President.....Van Ness Harwood
Secretary.....G. E. Kuhn
Treasurer.....M. C. Beman
Board of Governors: M. C. Beman, Roswell Farnham, D. J. Mahoney, J. M. Quackenbush, C. W. Stone

Wisconsin

Organized 1922
Headquarters, Milwaukee

President.....L. C. Plaehn
Vice President.....I. J. Rossiter
Secretary.....H. W. Alyea
Treasurer.....H. K. Forfar
Board of Governors: H. F. Brinen, A. A. Stutheit

Local Chapter Officers—1954 *(Concluded)*

Special Branch

Switzerland

Organized 1952
Headquarters, Zurich

President.....Walter Sennhauser
Vice President.....H. C. Bechtler
Secretary.....Walter Hausler
Treasurer.....John Frei

Student Branches

North Carolina State College

Organized 1948
Headquarters, Raleigh

President.....A. D. Ehrman
Vice President.....S. R. Moore
Secretary.....R. W. McDonald
Treasurer.....C. B. Wiggs
Reporter.....J. C. Deal

Oklahoma A. & M. College

Organized 1950
Headquarters, Stillwater

President.....R. P. Schmidt
Vice President.....C. E. Durrett
Secretary.....R. M. Jauch
Treasurer.....G. D. Crocker
Faculty Advisor.....R. R. Irwin

Oregon State College

Organized 1949
Headquarters, Corvallis

President.....W. E. Norman
Secretary.....P. I. Anderson
Treasurer.....R. L. Reiley
Publicity Chairman.....J. A. Stark

Purdue University

Organized 1949
Headquarters, W. Lafayette, Ind.

President.....E. B. Katz
Secretary.....R. A. Chambers

Texas A. & M. College

Organized 1946
Headquarters, College Station

President.....G. W. Millsap
Vice President.....W. E. Anderson
Secretary-Treasurer.....S. I. Perl

University of Detroit

Organized 1949
Headquarters, Detroit, Mich.

President.....J. C. Rumpf
Vice President.....J. T. Pillittere
Secretary-Treasurer.....L. J. Hayes, Jr.
Reporter.....V. W. Wiktorowski

University of Texas

Organized 1949
Headquarters, Austin

University of Toronto

Organized 1951
Headquarters, Toronto, Ont., Canada

President.....Alexander Tudor
Secretary.....T. M. Devine

TRANSACTIONS

of

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1493

SIXTIETH ANNUAL MEETING, 1954

HOUSTON, TEXAS

THE 60th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was held in Houston, Texas, January 25-27, 1954. Total registration broke all previous records for a non-exposition meeting with 1,102 registrants. There were 607 members, 272 ladies and 223 guests registered from 36 states, Canada, New Mexico and points as far distant as Stockholm, Sweden.

Four technical sessions were held in the Ballroom of the Rice Hotel, meeting headquarters. Other meeting highlights included an address by Houston's Mayor, Roy Hofheinz, at the Welcome Luncheon, a visit to the San Jacinto Monument and Battleship Texas followed by a Shore Dinner at the San Jacinto Inn and the 60th Annual Banquet on Wednesday night at which Dr. W. W. Kemmerer, Executive Director, Houston World's Fair, presented a talk *An Educator Looks at Air Conditioning.* Retiring Pres. Reg. F. Taylor received the Past-President's Medal from Past-President Merrill F. Blankin while Walter L. Fleisher, New York, N. Y., was honored for his achievements in the field of air-conditioning for human comfort by the award of the F. Paul Anderson Medal for 1953.

FIRST SESSION, MONDAY, JANUARY 25TH, 9:30 A.M.

The 60th Annual Meeting was called to order in the Ballroom of the Rice Hotel by Pres. Reg. F. Taylor who gave his report reviewing the growth of the Society and pointing out the progress made during his term of office. He noted that since his joining the Society in 1915 when it had 500 members and but 3 chapters, it had grown to over 9,300 members with 56 chapters, 9 student branches and a special branch in Zurich, Switzerland. He stated that the Society's membership growth has been paralleled by the growth of its publications. These include the JOURNAL, the HEATING VENTILATING AIR-CONDITIONING GUIDE, first published in 1922 with a circulation today of some 18,000 copies,

and the TRANSACTIONS started in 1895 and issued annually since that date. He outlined the founding in 1919 and the growth of the Society's Research Laboratory and some of its important accomplishments.

Mr. Taylor briefly summarized his chapter visits while an officer of the Society. The passing of five prominent Society members during 1953 was noted with regret; these include: Cyril Tasker, Dr. F. E. Giesecke, Dr. Homer Addams, Louis A. Harding and Prof. G. L. Larson.

TREASURER'S REPORT

The treasurer, John W. James, Chicago, Ill., presented the Treasurer's Report. This includes the accountant's report for the fiscal year ending October 31, 1953.

1952-1953 A.S.H.V.E. CONSOLIDATED CASH BUDGET COMPARISON

	Budget 12 Months	Actual 12 Months
INCOME:		
HEADQUARTERS:		
A—Dues (60 percent)	$122,670	$119,929.93
B—Publications	159,500	162,569.48
C—Investments	1,000	927.09
D—Other Income	2,400	3,131.50
	$285,570	$286,558.00
RESEARCH—E		
Dues	63,390	64,080.37
Contributions	70,000	46,616.37
Navy Contract	24,500	35,932.37
Interest	500	563.20
Transfer from Reserve	20,000	20,000.00
	$178,390	$167,192.31
TOTAL INCOME	$463,960	$453,750.31
EXPENSES:		
HEADQUARTERS:		
F—Committees and Chapters	$ 21,550	$ 21,220.90
G—Meetings	8,500	7,923.53
H—Publications	112,300	122,176.42*
I—Headquarters	128,650	128,707.98
K—Fund Raising	15,500	11,551.07
	$286,500	$291,579.90
RESEARCH—J		
Committee Expense	$ 5,500	$ 4,210.46
Staff Salaries	91,450	79,161.51
Laboratory Expenses	20,500	17,716.52
Building Operations & Maintenance	13,000	10,002.05**
Cooperative Research	29,200	22,025.66
Navy Contract	17,800	24,834.26
	$177,450	$157,950.46
TOTAL EXPENSES	$463,950	$449,530.36

* Deducted $11,619.55 from Publications chargeable to prior years TRANSACTIONS, and added **$2,000 chargeable to Building Maintenance.

Budgeted Excess	+ 10	+4,219.95
Special Appropriations:		
Membership Campaign	1,200	723.37
AMA Physiological Exhibit	1,500	1,500.00
Actual Excess	$-2,690	$+1,996.58

Respectfully submitted,
JOHN W. JAMES, *Treasurer*

Accountant's Report

FRANK G. TUSA & CO.
CERTIFIED PUBLIC ACCOUNTANTS
37 Wall St., New York 5, N. Y.

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
62 Worth Street,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y. and Cleveland and its related funds for the fiscal year ended October 31, 1953, and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1953. For the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements, and the disbursements were supported by payment vouchers. The dues from members and the interest from investments were accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business October 31, 1953, is submitted herewith and your attention is directed to the following comments thereon:

CASH

The Cash on Deposit was verified by direct communication with commercial and savings banks and the balances reported to us were reconciled with those reflected by the books of the Society. A schedule of cash is included as a part of this report.

Checks representing the cash on hand for deposit were inspected by us and the cash on hand was counted.

MARKETABLE SECURITIES

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Company, where same are deposited for safe-keeping. Securities have been included in the accompanying balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

ACCOUNTS RECEIVABLE

Trial balances taken of the membership dues receivable and sundry debtors as of the close of business October 31, 1953, were classified and aged as follows:

MEMBERSHIP	Prior	1953	Total
Members	—0—	$ 3,206.25	$ 3,206.25
Associates	$50.48	3,218.75	3,269.23
Affiliates	—0—	1,979.25	1,979.25
Juniors	—0—	1,604.83	1,604.83
Students	1.00	18.00	19.00
TOTAL	$51.48	$10,027.08	$10,078.56

SUNDRY DEBTORS

Unpaid charges made during 1953	$5,600.01
Unpaid charges made during 1952	437.42
TOTAL	$6,037.43

After writing off all receivables known to be uncollectible, it is our opinion that reserves for dues and accounts receivable doubtful of collection reflected in the accompanying Balance Sheet are ample to cover collection losses that may be incurred.

RESEARCH FUND ACCOUNTS RECEIVABLE

The balance due to the Research Fund from the Society as at October 31, 1953, represents the unpaid balance of 40 percent of the dues receivable from members, associates, and affiliates as of such date.

Included in the miscellaneous accounts receivable of $303.07 is an amount due from the Estate of Cyril Tasker for $295.69. This represents travel advances of $450.00 not accounted for to Research, less an adjustment of $154.31 due to the Estate for amounts withheld on account of insurance premiums advanced to the Pension Trust for the account of Mr. Tasker.

INVENTORIES

The following inventories were verified either by physical count or direct communications.

TRANSACTIONS	$ 8,644.49
Emblems	585.15
GUIDE PAPER	7,831.60
Membership Roll Paper	200.00
Membership Roll Books	2,214.00
TOTAL	$19,475.24

A schedule of TRANSACTIONS inventoried follows:

Volume	Year	Quantity	Price	Amount
1-51	1895-1945	3,616	.40	$1,446.40
52	1946	404	1.67	674.68
53	1947	161	1.77	284.97
54	1948	787	1.78	1,400.86
55	1949	278	2.02	561.56
56	1950	705	2.08	1,466.40
57	1951	1,235	2.275	2,809.62
TOTAL				$8,644.49

DEPOSITS RECEIVABLE

The deposit placed with the United Airlines in the sum of $425.00 was verified by direct communication.

ADVANCES

The indebtedness from employees in New York and Cleveland represents advances to the employees retirement plan in the sums of $1,352.54 and $835.90 respectively. This indebtedness representing one years advance premiums is to be reimbursed by the trust during the ensuing fiscal year.

DEFERRED CHARGES

The following prepayments have been deferred to future operations:

H.P.A.C. Subscriptions	$3,008.61
Insurance Premiums—Cleveland	1,977.74
Insurance Premiums—New York	182.86
TRANSACTIONS—Volume 59	14.69
TOTAL	$5,183.90

PERMANENT ASSETS

The Land and Buildings, Instruments, Equipment and Furniture and Fixtures are reflected on the Balance Sheet at cost of acquisition. With the exception of Land and Buildings all assets have been depreciated at the rate of 10 percent per annum. During the year fully depreciated Furniture and Fixtures in the amount of $219.68 was written off against the reserve.

In accordance with the resolution adopted by the Council at its meeting of January 23, 1949, depreciation was not provided on the Buildings for the current fiscal year.

TAXES

The sum of $1,526.00 represents Federal Income Taxes withheld from employees salaries during the month of October, 1953.

DEFERRED INCOME

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follows:

MEMBERSHIP	AMOUNT
Members	$ 580.44
Associates	227.37
Affiliates	438.25
Juniors	558.52
Students	19.50
TOTAL	$1,824.08

RESERVE FOR TRANSACTIONS

Out of the original $13,500.00 provided for TRANSACTIONS Volume 58 there remains an unexpended balance of $13,476.25, which is reflected in the accompanying Balance Sheet.

RESERVE FOR BUILDING MAINTENANCE

On October 31, 1952, the Council voted:

"That disbursements be made from the Building and Maintenance Reserve Funds authorized by the Building Committee for major items of repair and maintenance, but any balance remaining in the fund beyond the expenditures for each year would go into reserve until the Building Maintenance Reserve Fund reached a total of $10,000.00."

Inasmuch as there were no disbursements authorized as at the end of the current fiscal year, there exists a reserve balance of $4,000.00 as at October 31, 1953.

FUNDS

There is included as a part of this report an analysis of funds reflecting the changes that occurred therein during the fiscal year ended October 31, 1953.

INSURANCE

The insurance coverages of the Society follow:

FIRE		
Building (Cleveland)		$201,000.00
Personal Property:		
Cleveland	$104,000.00	
New York	25,500.00	
Printers	12,000.00	141,500.00
SPRINKLER LEAKAGE		
Waverly Press	10,000.00	
Hawes and Petit	2,000.00	12,000.00
NON-OWNERSHIP—AUTOMOBILE—COMPREHENSIVE		
Public Liability		100/300M
Property Damage		25M
Medical		1M

GENERAL PUBLIC LIABILITY—PREMISES	
New York and Cleveland	
Liability	100/300M
Bodily Injury	25/100M
COMMERCIAL BLANKET BOND	
All employees including the President, Treasurer and Chairman of the Finance Committee	20,000.00

Respectfully submitted,

FRANK G. TUSA & CO.
Certified Public Accountants

BALANCE SHEET

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y.—October 31, 1953

ASSETS

GENERAL FUNDS				
SOCIETY ASSETS				
CASH				
On Hand and on Deposit			$57,351.52	
SECURITIES				
At Cost (Market Value $3,680.00)		$3,782.36		
Add: Accrued Interest		27.50	3,809.86	
ACCOUNTS RECEIVABLE				
Membership Dues	$10,078.56			
Less: Reserve for Doubtful	6,047.13	4,031.43		
Advertisers and Sundry Debtors	6,037.43			
Less: Reserve for Doubtful	1,000.00	5,037.43		
Initiation Fees		80.00	9,148.86	
INVENTORIES			19,475.24	
DEPOSIT WITH UNITED AIRLINES			425.00	
ADVANCES TO EMPLOYEES PENSION TRUST			1,352.54	
DEFERRED CHARGES			3,206.16	
TOTAL SOCIETY GENERAL FUND ASSETS				$94,769.18
RESEARCH GENERAL FUND ASSETS				
CASH				
On Hand and on Deposit			36,136.42	
ACCOUNTS RECEIVABLE				
40% of Dues	4,497.14			
Less: Reserve for Doubtful	2,029.14	2,468.00		
Due from U. S. Navy Dept.		3,903.91		
Hospitalization—Employees		131.95		
Miscellaneous		303.70	6,807.56	
ADVANCES TO EMPLOYEES PENSION TRUST			835.90	
DEFERRED CHARGES			1,977.74	
TOTAL RESEARCH GENERAL FUND ASSETS				$45,757.62
PROPERTY FUND				
Land and Buildings		$84,920.96		
Less: Reserve for Depreciation		3,908.07	$81,012.89	

PROPERTY FUND (*continued*)

Laboratory and Equipment (Cleveland)	65,744.42		
Less: Reserve for Depreciation	33,221.28	32,523.14	
Furniture and Fixtures (New York)	14,133.54		
Less: Reserve for Depreciation	4,867.81	9,265.73	
Library (Cleveland)		300.00	
Tools (Cleveland)		300.00	123,401.76
BUILDING FUND			
CASH			
On Hand and On Deposit		830.48	
SECURITIES			
At Cost (Market Value $36,719.50)	32,519.00		
Add: Accumulated and Accrued Interest	4,170.50	36,719.50	37,549.98
RESERVE FUND			
Cash on Deposit		20,343.02	
Securities at Cost (Market Value $66,659.00)	53,631.50		
Add: Accumulated Interest	13,027.50	66,659.00	87,002.02
ENDOWMENT FUNDS			
F. PAUL ANDERSON FUND			
Cash on Deposit		226.21	
Securities at Cost (Market Value $973.00)	1,000.00		
Add: Accrued Interest	12.50	1,012.50	1,238.71
RESEARCH FUND			
Cash on Deposit			721.85
RESEARCH RESERVE FUND			
Cash on Deposit			32,102.74
TOTAL ASSETS			$422,513.86

LIABILITIES

GENERAL FUND				
FEDERAL WITHHOLDING TAXES			$ 1,526.00	
DUE RESEARCH			2,468.00	
ACCRUED ACCOUNTS				
Salaries and Commissions		$11,268.24		
Professional Fees		1,000.00	12,268.24	
SUNDRY ACCOUNT PAYABLE			127.55	
DEFERRED INCOME				
Prepaid Membership Dues				
Elected Members	$1,824.08			
Candidates	1,012.50	2,836.58		
Prepaid Admission Fees		2,039.50	4,936.08	
RESERVE FOR PUBLICATIONS				
TRANSACTIONS—Volume 58			13,476.25	
RESERVE FOR FLUCTUATION IN CANADIAN EXCHANGE			33.26	$ 34,835.38
RESEARCH FUND				
RESERVE FOR BUILDING MAINTENANCE			4,000.00	

RESEARCH FUND (*continued*)			
DEFERRED INCOME			
Air Cleaning	2,179.00		
Sound Energy Studies	574.39		
Summer Infiltration	2,500.00		
Sorption	500.00	5,753.39	9,753.39
NET WORTH			
Society General Fund		$ 50,343.80	
Research General Fund		35,904.23	
Property Fund		123,401.76	
Building Fund		37,549.98	
Society Reserve Fund		96,692.02	
F. Paul Anderson Fund		1,238.71	
Research Endowment Fund		721.85	
Research Reserve Fund		32,102.74	377,955.09
TOTAL LIABILITIES AND NET WORTH			$422,543.86

COMPARATIVE STATEMENT OF INCOME AND EXPENSES

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—NEW YORK, N. Y.

For the Fiscal Year Ended October 31, 1953, and October 31, 1952

	Fiscal Year Ended October 31, 1953	October 31, 1952	Increases (Decreases)
INCOME			
Income from Dues	$123,800.89	$113,564.65	$ 10,236.24
Admission Fees	9,690.00	7,243.00	2,447.00
Emblems, Pins, Etc.	1,172.00	626.88	545.12
PUBLICATIONS			
JOURNAL Contracts	23,500.00	23,500.00	
GUIDE	132,973.58	126,846.41	6,127.17
TRANSACTIONS	3,158.79	1,159.25	1,999.54
Books, Reprints, Codes, Etc.	3,446.80	3,016.95	429.85
Income from Investments	927.09	905.86	21.23
RESEARCH			
Dues	64,381.43	60,065.60	4,315.83
U. S. Navy Research	34,798.45	27,061.95	7,736.50
Contributions—General	17,535.37	19,272.94	(1,737.57)
Contributions—Earmarked	29,852.00	42,971.84	(13,119.84)
Interest	39.26	1.48	37.78
TOTAL INCOME	$445,275.66	$426,236.81	$ 19,038.85
EXPENSES			
Committees and Chapters	$ 20,995.59	$ 21,321.79	(326.20)
Meetings	5,923.53	5,001.30	922.23
PUBLICATIONS			
Members Subscriptions to H.P.A.C.	16,618.29	15,876.54	741.75
TRANSACTIONS	21,737.45	15,006.26	6,731.19
Membership Roll	2,918.57	8,888.08	(5,969.51)
Books, Reprints, Codes, Etc.	2,510.91	2,499.54	11.37
GUIDE	83,128.64	81,201.60	1,927.04
Headquarters	128,687.05	114,270.99	14,416.06
Fund Raising	11,551.07	13,734.89	(2,183.82)

EXPENSES (*continued*)

RESEARCH			
Committees Expenses	4,210.46	4,269.59	(59.13)
Staff Salaries	75,411.51	80,771.59	(5,360.08)
Laboratory Expenses	18,340.33	22,004.39	(3,664.06)
Building Operation and Maintenance	9,454.06	9,512.33	(58.27)
Provision for Building Maintenance Reserve	2,000.00	2,000.00	
U. S. Navy Research	24,834.26	17,839.83	6,994.43
Cooperative Research	22,025.66	28,454.52	(6,428.86)
Other Deductions	16,391.93	9,446.84	6,945.09
TOTAL EXPENSES	466,739.31	452,100.08	14,639.23
EXCESS OF INCOME OVER EXPENSES	(21,463.65)	(25,863.27)	(4,399.62)

REPORT OF COUNCIL

As required by Section 46 of the Membership Corporations' Law of the State of New York the following report is presented by the Council and filed herewith:

Assets

The Society has the following Assets:

Land and Laboratory Buildings at Cleveland, Ohio	$ 81,012.89
Furniture and Equipment, Tools, Etc., at New York and Cleveland	42,388.87
On Deposit in New York and Cleveland Banks	147,712.24
Securities (United States and Utility Bonds) with Bankers Trust Company, New York as Custodian	108,200.86
Accounts Receivable	13,052.77
Inventory Items	19,475.24
Deferred Items	5,183.90
Miscellaneous	3,049.09
TOTAL	$420,075.86
1952 TOTAL	384,920.41
INCREASE	$ 35,155.45

The sources of Society Cash Receipts were:

Admission Fees and Dues	$134,662.89
Publications	163,079.17
Interest	927.09
From Research	146,606.51
TOTAL	$445,275.66

Cash Disbursements were:

Meetings, Committees and Chapters	$26,919.12
Publications	126,913.86
Headquarters	128,687.05
Fund Raising	11,551.07
Research	160,026.28
TOTAL	$454,097.38

The Society's Reserve Fund was increased by the addition of $9,690.00 Admission Fees and $2,559.13 Interest earned on Savings, Bank Accounts and Securities. This is an increase of $12,249.13, bringing the Reserve Fund to $96,692.02.

Now in the Custodian Account of the Bankers Trust Co., New York, there are securities of the United States and Canadian Governments and a utility which cost: General Fund $3,782.36, Building Fund $32,549.00, Reserve Fund $53,631.50, and F. Paul Anderson Fund $1,000.00. The Research Reserve Fund now amounts to $32,102.74.

Liabilities

The Society has the following Liabilities:

Reserve for Building Maintenance	$ 4,000.00
Accounts Payable	127.55
Federal Withholding Taxes	1,526.00
Accrued Accounts	12,268.24
Deferred Income:	
Research Projects	5,753.39
Prepaid Dues, Initiation Fees, Etc.	4,936.08
Reserve for	
TRANSACTIONS	13,476.25
Fluctuation in Canadian Exchange	33.26
Net Worth (Funds)	377,955.09
TOTAL	$420,075.86

At the end of the fiscal year, October 31, 1953, the Balance Sheet shows the Society's Net Worth as $377,955.09 compared to $343,990.17 on October 31, 1952, a net gain of $33,964.92.

The insurance coverage of the Society is as follows: Fire for Buildings $201,000.00, for contents $41,500.00; Sprinkler $12,000.00; Automobile: Comprehensive Public Liability $100,000-$300,000.00, Property Damage $25,000.00, Medical $1,000.00; General Public Liability—Society Premises: Liability $100,000-$300,000.00, Bodily Injury $25,000-$100,000.00; Fidelity Bonds on Officers and all employees $20,000.00. Workmen's Compensation is carried in New York and Ohio, as required by law.

The Society has contributed to the Employees Retirement Plan Trust $2,417.21 and the participating employees contributed a similar amount deducted from salary.

Total cash in New York and Cleveland banks $147,712.24, consisting of $57,351.52 in General Operating Fund, $20,343.02 in Reserve Fund, $226.21 in F. Paul Anderson Fund, $36,136.42 in the Research Funds, $32,102.74 in Research Reserve, $721.85 in Research Endowment, and $830.48 in the Building Fund.

Membership

The Society admitted the following members in the grades indicated since January 1, 1953:

Members	329
Associates	11
Affiliates	446
Junior Members	393
Students	117
TOTAL	1,296

The names and addresses of the candidates for membership were published in the JOURNAL of the Society each month during the year and are on record in the Secretary's office. The present membership total is 9298.

Respectfully submitted,

REG. F. TAYLOR, *President*
JOHN W. JAMES, *Treasurer*

President Taylor announced that the next item of business would be consideration of the Amendments to the By-Laws which had been introduced for discussion at the Semi-Annual Meeting, 1953, Denver, Colorado.

In accordance with the By-Laws, all members had been notified that a vote on the amendments would be taken at this meeting. First Vice Pres. L. N. Hunter presented the following resolution which was seconded and approved unanimously:

Resolved, That *Article VII, Section 2 (e)* of the Society's By-Laws be deleted and *Be it further Resolved,* That *Article VII, Section 3* of the By-Laws be amended by adding thereto a new subdivision *(j)* reading as follows:

(j) Standards Committee, consisting of six (6) Members. The said committee shall consider all scientific questions and data pertaining to engineering codes and standards, initiate, and propose changes and improvements thereof for the public benefit, and report its recommendations to the Council.

J. Donald Kroeker, Portland, Ore., moved the adoption of the following resolution which was seconded and also approved unanimously:

Resolved, That the last sentence of *Article VIII, Section 7,* of the By-Laws be amended to read as follows:

. . . The elected candidates shall be installed during the Annual Meeting and their terms shall commence at the close of the last session of the Annual Meeting.

The President appointed the following tellers of election: S. W. Beaty, Shreveport, La., *Chairman;* A. B. Ullrich, Jr., and R. E. Haubold, Houston, Tex. The session was then turned over to L. T. Mart, Kansas City, Mo., who acted as chairman.

Three technical papers were presented and discussed (see Program, page 23).

SECOND TECHNICAL SESSION, TUESDAY, JANUARY 26TH, 9:30 A.M.

The second technical session was called to order at 9:30 a.m. by First Vice President L. N. Hunter who introduced R. S. Dill, Chairman of the Committee on Research who presented the Committee's Annual Report.

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1953

THIS REPORT is perhaps unique among Reports of Committees on Research in that it contains an account of some difficulties; these must be recorded and considered. First, it was found necessary to reduce the research budget this fiscal year; second, during the year, the Laboratory has lost personnel which we have been unable to replace; and last, but not least, we lost the Director of Research, Cyril Tasker, by death from a heart attack on May 27. Mr. Tasker was a man of energy and ability, and it is possible that our other difficulties could have been avoided or at least ameliorated if his talents, and particularly his knowledge of research affairs, had continued to be available to us. C. M. Humphreys was appointed Acting Director of Research, by Council, at its July meeting, and his experience at the Laboratory was undoubtedly beneficial in carrying forward our program. The Laboratory and cooperative projects have therefore produced data and papers according to their intended functions and some details worthy of note are set forth below.

Budget and Finances: The research budget for the last fiscal year (ending October 31) was $178,390.

Actual expenditures were less than the budget chiefly because manpower was not available to activate some projects upon which the budget was based. Both the staff

and the annual budget should be increased. The matter of fund raising is now under serious consideration by the Ways and Means Committee, and it is probable that it will offer Council some recommendations to this end at the Houston meeting. A possible reason for lack of ample funds for all purposes may be that the Society's activities under the Committee on Research have not been made well enough known to the membership and to the sponsors of projects. The Committee on Research and the Technical Advisory Committees are known to be manned by capable and willing individuals and the merit of their actions is apparent to anyone attending their meetings or listening to their discussions. An urgent need, therefore, is for better dissemination of information which may be achieved by more frequent progress reports and an improvement in research papers in regard to clarity of conclusions and applicability of results. To achieve this, it is fairly clear that additional literary ability must somehow be added to the laboratory staff, since the present staff is not large enough to conduct its work and also produce the necessary reports and papers. A recommendation that a competent technical writer be added to the staff has been on record for several years, and it is to be hoped that competent help of this kind can be attracted to our service soon. There is strong argument that the duty of the Director of Research should be largely that of an editor because on our papers depend the future and fate of the Research Laboratory and of the research activity.

Laboratory Staff: The laboratory staff has lost a number of members in the last several years. It is apparent that those remaining have considerable competence in their various fields and a desirable loyalty to A.S.H.V.E., but they are too few in number.

As stated, this staff should be augmented and, certainly, further losses should be avoided. Technical writing ability now exists among present staff members and should be given opportunity to flourish. Additional help should also be obtained to relieve anyone assigned to writing.

Lack of adequate staff is now hampering authorized work. In particular the projects on odors, sensations of comfort, and others are suffering from this cause. There is reason to believe that our salary scale is not high enough to attract new graduates from technical schools or colleges and, at its meeting on January 4, the Research Executive Committee approved Mr. Humphreys' suggestion that he survey the situation in further detail and prepare some recommendations for consideration by the Committee on Research at the January meeting in Houston.

Research Policy and Planning: The Operational Guide for the Committee on Research has been approved and is now in force so that the functions of the Committee, the Director of Research and the laboratory staff are better defined than heretofore. There is now a Council Committee on Long Range Planning, which will submit some recommendations during the Houston meeting, and this is expected to yield further guidance of the research program. The Committee on Research has had a Long Range Research Program for several years and this program is now under review by a subcommittee under the chairmanship of Prof. B. H. Jennings. The research activity of the Society is therefore under examination from several directions and we can hope that the serious consideration now being given it will render the program even more valuable than heretofore.

Research Projects: During the fiscal year ending last October, there were 10 active projects at the Research Laboratory at Cleveland and 16 cooperative research projects at various institutions over the country. These projects are enumerated in the Report of the Acting Director of Research in his discussion of Technical Advisory Committee activities. In this report, some details are given on a few of the numerous accomplishments as follows:

On the subject of radiant heating, the output of floor panels and the resultant room air temperatures were established for steady state conditions and for several different infiltration rates. Experimental work was done in the Environmental Laboratory at Cleveland, and can be expected to go far toward settling long-standing controversies

on this subject. On ceiling panels, information was gathered on the effect of good *vs.* poor construction and the importance of good contact between tubes and lath was shown. The data will be prepared soon for presentation in THE GUIDE and for use by manufacturers in design manuals.

On window heat gain, data were obtained on the solar heat gain through windows with several shading devices of the venetian-blind type. These data are expected to appear in the next issue of THE GUIDE and should be of much interest to designers since shading is one important means of reducing required air conditioning capacity for buildings. The work was done in the solar calorimeter at the Cleveland Laboratory under the supervision of G. V. Parmelee, and additional work and a follow-up paper are in prospect giving the relation between radiation and convection components of the heat released from these shading devices.

On thermal insulation, the Laboratory has assisted the Technical Advisory Committee concerned through the efforts of M. L. Erickson and, later, D. J. Vild. By means of questionnaires sent to various laboratories, data are being gathered for revising a chapter on Thermal Insulation in THE GUIDE, with the purpose of publishing heat transfer coefficients for the various materials. Rather than citing data from a single source for a given material the objective is to publish acceptable values based on laboratory tests and weighted in accord with the Committee's judgment. M. W. Keyes' TAC on Insulation will deserve the thanks of the Society and of the profession for this accomplishment.

On the superheating of water, the phenomenon has been demonstrated in glass vessels at Northwestern University by Prof. B. H. Jennings and his associates as part of the project under the auspices of John W. James' TAC on Hot Water and Steam Heating. Experiments to investigate the effect in conventional boilers are in prospect.

The various projects are dealt with in detail in the report of the Acting Director of Research.

A.S.H.V.E. RESEARCH IN COOPERATING INSTITUTIONS

California Institute of Technology, Pasadena, Calif.: (Research Fellowship) The Role of Air Conditions in Plant Growth, Plant Reproduction and Photosynthesis.

University of California, Los Angeles, Calif.: Thermal Circuit Techniques Applied to Heating and Cooling Problems.

Case Institute of Technology, Cleveland, Ohio: (Research Fellowship) Air Flow from Annular Slots, Ceiling Plaques and Circular Diffusers.

Columbia University, New York, N. Y.: The Investigation of Thermally Actuated Moisture Migration in Granular Media (Heat Pump).

Cornell University, Ithaca, N. Y.: The Sol-Air Thermometer and the Sol-Air Temperature Concept.

University of Florida, Gainesville, Fla.: Air Entrainment in Hot Water Heating Systems.

University of Illinois, College of Engineering, Urbana, Ill.: Study of Infiltration in a Residence Using the Tracer Gas Technique.

University of Illinois, College of Medicine, Chicago, Ill.: The Physiological Adjustments made by Healthy and Physically Impaired Persons to Rapid Changes in Atmospheric Environment.

Kansas State College, Manhattan, Kan.: The Downward Projection of Heated Air Streams.

Michigan State College, East Lansing, Mich.: Friction in Rectangular Branch Take-Off Fittings.

University of Minnesota, Minneapolis, Minn.: (a) Solar Energy as a Potential Heat Source for Heat Pump Installations. (b) (Research Fellowship)—Field Studies on the Control of Panel Heating Systems.

Northwestern University, Evanston, Ill.: (a) Noise in Piping Systems. (b) Metastable State of Water as a Possible Explosion Hazard in Heating Boilers.

The Pennsylvania State College, State College, Pa.: (Research Fellowship)—The Water Vapor Permeance of Building Materials.

University of Pittsburgh, Pittsburgh, Pa.: (Research Fellowship)—Design Requirements of Local Exhaust Ventilation for Control of Atmospheric Contaminants Released by Hot Industrial Processes.

During the Society year now commencing the obvious immediate objectives of the research program include the installation of a permanent Director of Research. It is not yet certain that all worthy candidates have been considered for nomination so that the Committee on Research may propose a further postponement to assure all interested parties an opportunity to present their cases.

The staff at the Laboratory should be augmented immediately if possible and the salary scale and other contributing factors must be adjusted to this end.

The arrangement for producing and editing papers must be formulated and put into operation.

The effort to attain closer liaison and cooperation between the Technical Advisory Committees, the Committee on Research, the Laboratory and the cooperating institutions must be continued. The short term or present program must be prosecuted as effectively as possible and due consideration must be given to the long-range program.

Research Papers: The Research Program of the A.S.H.V.E. yielded 15 papers for presentation at the 1953 Semi-Annual Meeting and the 60th Annual Meeting. Eight of these appeared in the A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, and the balance are presented at Houston and appear in this volume of the TRANSACTIONS.

REPORT OF ACTING DIRECTOR OF RESEARCH

The Chairman of the Committee on Research has mentioned in his report the great loss suffered by the Laboratory last May with the passing of Mr. Tasker. Since that time every effort has been made to carry on the work of the Laboratory in accordance with the principles which he had established.

Staff: During 1953, the Research Laboratory, like other research organizations and industries throughout the country, found it increasingly difficult to obtain enough trained personnel to maintain its planned research schedule. Every effort is being made to increase the Laboratory staff.

Equipment and Facilities: Equipment and facilities at the Research Laboratory continue to increase with each passing year. Although no major piece of equipment was added during 1953, important additions to instrumentation and equipment were made.

It is imperative that major work soon be undertaken at the Laboratory in the field of periodic heat flow. Considerable thought has been given to the type of computing apparatus or analogue equipment best suited for such work. It is hoped that definite steps may be taken to provide such equipment for the Laboratory in 1954.

Guide: Development of fundamental data valuable to the heating and air conditioning industry is the ultimate goal of all Society research. It is not surprising, therefore, that new data, evolved under the Society's research program are found in THE GUIDE each year. THE GUIDE 1954 will contain Research Laboratory data on the effect of slat-type shades in reducing solar heat gain through sunlit glass.

During the past year, several Technical Advisory Committees have taken a very active part in the task of reviewing and revising THE GUIDE.

AIR DISTRIBUTION AND AIR FLOW

During 1953, three cooperative research projects have been carried on under the guidance of the TAC on Air Distribution.

Air Distribution Studies (Case Institute of Technology): Cooperative research on air distribution, in progress at Case Institute of Technology for many years, has been the subject of a number of papers presented before the Society. At the present time, two separate studies are under way. One is concerned with the performance of radial jets of heated or cooled air discharged at various angles; the second is to determine the laws governing the intermixing of two or more jets, the natural streams of which tend to overlap.

Work on these projects has involved study of related problems and obtained interesting results. A paper, Computing Temperatures and Velocities in Vertical Jets of Hot or Cold Air, by Alfred Koestel, has been submitted and will be published in the near future. The treatise presents an analytical solution for performance of a jet of air discharged vertically upward or downward.

Downward Projection of Heated Air (Kansas State College): Work on this cooperative research project continued throughout 1953 with the active support of the *Industrial Unit Heater Association.* Original work was done with jets discharging vertically downward from standard *A.S.M.E.* nozzles. Tests were then made with a unit heater, and the results correlated.

As a by-product of this investigation, an electronic pressure-differential recorder has been developed which will permit the automatic recording of velocity measurements made with a pitot tube. This instrument will greatly facilitate further work on this project.

Friction in Rectangular Take-Off Fittings (Michigan State College): A study has been made of the frictional resistance of three different types of rectangular branch take-off fittings. Each fitting was tested with various percentages of the total flow through the continuing trunk and its branch outlets. Pressure losses in both streams were determined. The experimental phase has been completed and a report on results of the work is being prepared.

AIR CLEANING

At the meeting of the TAC on Air Cleaning held in January 1953, it was decided not to submit the third draft of the proposed A.S.H.V.E. filter test code to the Committee on Research. Since that time, there has been no further work on the development of a Society code on air cleaning.

The Committee on Research was asked by the *Air Filter Institute* to recommend that the 1933 A.S.H.V.E. Filter Test Code be rescinded, and to recognize and approve the *A.F.I.* test code, but no action was taken.

COMBUSTION

Some time ago, in presenting its long-range program to the Committee on Research, the TAC on Combustion indicated that problems associated with pulsation in oil flames and resonance in gas flames merited high priority. A number of proposals for cooperative research on these subjects have been received and are being considered.

Financing of the work has recently been arranged through agreements with *Oil Heat Institute* and the *American Gas Association,* under which each group will contribute one third of the required funds.

COOLING LOADS

All activities under the guidance of the TAC on Cooling Load during 1953 were pointed toward development of more accurate methods for determining instantaneous equipment loads. Studies on this subject are being made at the Society Laboratory and at cooperating institutions.

Air Conditioning Thermal Circuit (A.S.H.V.E. Research Laboratory): In 1951, H. B. Nottage developed the thermal circuit concept to portray and interrelate factors which influence the instantaneous equipment load. To test the feasibility of this method of approach, equations were written for the heat gain to a single room with northern exposure, in one case with a large glass block panel, and in another, with an ordinary window glass. In both cases, direct radiation through the glass was neglected. The equations were solved with the aid of calculating equipment at Case Institute of Technology. A comparison of the results thus obtained, with those calculated by the method now outlined in THE GUIDE, which does not consider the heat capacity of the internal structure, demonstrates the importance of considering the entire thermal system. A report on this work was presented at the Houston meeting.

As a second phase of this problem, equations for the same room have been rewritten to include radiation directly transmitted through glass. As before, solutions were obtained by computing equipment and are now being subjected to further analysis.

Extension of this initial effort in analysis of complete thermal systems involving periodic heat flow is highly desirable. Thermal circuit analysis can make important contributions in a variety of problems in the field of heating, ventilating, and air conditioning. Further work in this direction is being considered.

Periodic Heat Flow Through Roof Sections (A.S.H.V.E. Research Laboratory): During the summer of 1953, experiments were conducted to determine the periodic

heat flow through roofs. Two roof sections, one of heavy and one of light construction, were used. The heavy section consisted of 3-in. concrete, 2-in. rigid insulation, and built-up roofing. The lighter section consisted of a 3-in. precast calcium silicate slab to which build-up roofing was applied.

Test sections were mounted horizontally in the calorimeter previously used for the glass studies. Calorimeter surface below the slab was maintained at a constant temperature while the top of the slab was exposed to the weather for test periods of at least 36 hr. Heat flow meters on top and bottom surfaces of the roofs measured heat entering or leaving the surfaces. The heavy roof section was tested, first with the insulation on top, then with the concrete on top.

At the present time, the experimentally determined heat transfer rates are being compared with results obtained by application of the theory for multi-layer construction proposed by C. O. Mackey and D. K. Wright, Jr.

Thermal Circuit Techniques Applied to Heating and Cooling Problems (University of California): The first phase of this new cooperative study of thermal circuit techniques applied to heating and cooling problems is prediction of the thermal environment inside a one room test house by analyzing the thermal circuit representing the heat transfer system. This analysis is being done with an equivalent resistance-capacitance electrical network and appropriate input devices to represent the boundary conditions. The elements of this network are now being developed and assembled.

Results of analogue solutions will be checked against experimental data, obtained for the test house on several clear days last September.

Joint I.E.S.-A.S.H.V.E. Committee: During 1953 a joint *I.E.S.*-A.S.H.V.E. Committee was formed to consider matters of common interest to the two groups. The committee, which consists of three members appointed by each society, has held two meetings. One of the aims of the Committee is to prepare suitable authoritative data on subjects of mutual interest for THE GUIDE and the *I.E.S. Lighting Handbook.*

EVAPORATIVE COOLING

During 1953, a new Technical Advisory Committee on Evaporative Cooling was organized under the chairmanship of A. J. Hess. The Committee is broken down into three sub-committees on Cooling Towers, Evaporative Condensers, and Evaporative Air Cooling.

The Committee is now considering a proposal for cooperative research.

HEAT FLOW THROUGH GLASS

All experimental work on the shading of sunlit glass was completed in 1952 but analytical work on this subject has continued throughout 1953. The results were presented in the form of practical design data at the Denver meeting in the paper, Design Data for Slat Type Sun Shades for Use in Load Estimating, by G. V. Parmelee and D. J. Vild.

Another paper, Convection and Radiation Gain from Windows with Slat-Type Sun Shades, is being prepared and should be ready for presentation at the 1954 Semi-Annual meeting.

Some analysis has been made of solar radiation data, taken at the Laboratory during the last seven years. Papers on this, and on two other secondary phases of the studies on heat flow through glass, will be prepared during 1954.

HEAT PUMP

Moisture Migration in Granular Media (Columbia University): To develop information on the heat absorbed by ground coils, a radioactive tracer technique was used to study moisture migration in a simulated soil made up of small glass beads. Experimental work, performed under a cooperative research agreement with the Society, was completed in 1952, and a paper, Moisture Movement in Soils Due to Temperature Difference, by W. A. Hadley and Raymond Eisenstadt, was presented at the Semi-Annual Meeting in Denver.

Solar Energy as a Heat Source for the Heat Pump (University of Minnesota): A study of the feasibility of using solar energy as a heat source for the heat pump has been in progress at the University of Minnesota since 1952 under a cooperative

research agreement with the Society. The investigation by R. C. Jordan and J. L. Threlkeld has been completed and the results were presented at the 1954 Annual Meeting.

The first part of the paper, Solar Energy Availability for Heating in the United States, deals with availability of solar energy during cold winter months in various parts of the country.

The second part, Utilization of Solar Energy for House Heating, discusses solar collectors and presents a rational method for calculating their efficiency. Storage of solar energy is considered and different methods compared.

The last part, Design and Economics of Solar Energy Heat Pump Systems, considers some of the more practical problems involved in the design of solar collectors and heat pump systems. A comparison of operating costs for heat pump and gas, oil, and coal-fired heating systems is given for four cities.

HEATING LOAD

Suitability of Helium as a Tracer Gas for Infiltration Studies (U. S. Bureau of Standards): Some time ago it was proposed that infiltration be studied by releasing a small amount of helium in a space and measuring its rate of decay. A gas analyzer for determining the concentration of helium was purchased by the Laboratory, but the TAC on Heating Load suggested that the feasibility of both method and instrument be demonstrated under carefully controlled conditions before field tests were attempted. The analyzer was therefore sent to the National Bureau of Standards where it was used for studying infiltration in the Test Bungalow.

Tests completed early in 1953 indicated that satisfactory results could be obtained with the instrument, and a complete report on its use was submitted. The instrument was then shipped to the University of Illinois for use in field testing.

Study of Infiltration in a Residence Using the Tracer Gas Technique (University of Illinois): Under a new cooperative research agreement with the University of Illinois, a study of infiltration is being made in the Warm Air Research Residence No. 2. Infiltration rates will be determined by the tracer gas technique, using a helium analyzer loaned to the school by the Research Laboratory. Tests will be made under a variety of wind and weather conditions.

Guide Revisions: The TAC on Heating Load prepared revisions for Chapter 12 of THE GUIDE 1954.

HOT WATER AND STEAM HEATING

Noise in Piping Systems (Northwestern University): This cooperative research project was activated in 1951. A paper, Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems, by W. L. Rogers, describing the experimental apparatus and instrumentation was presented at the Semi-Annual Meeting in Denver in 1953.

To date, all tests have been made with 20-ft test lengths of ¾-in. pipe. Other pipe sizes and lengths will be used, with water velocity, temperature and pressure as additional variables.

The Metastable State of Water as a Possible Cause of Boiler Explosions (Northwestern University): Initial work on this project was concerned with demonstrating the metastable state in as many ways as possible. It was easy to show its existence in glass and, by a gradual reduction in pressure, it could be produced in a metal container. Although normal conditions re-established themselves with considerable violence, the pressure impulses in the small vessels used up to now have been of small magnitude.

A larger tank is now being prepared to determine if the pressure surges will increase with an increase in the size of the vessel. If this is found to be the case, tests will be made in actual boilers.

Air Entrainment in Hot Water Heating Systems (University of Florida): Under a cooperative research agreement with the University of Florida, studies are being made on the effects of air or gases on the operation of hot water heating systems. The relationship between water velocity and air entrainment, and methods of air elimination are the subjects currently being investigated.

HUMAN CALORIMETRY

In 1948 the Office of Naval Research requested the Society's Laboratory to attempt to design and construct a human calorimeter which would have high accuracy and rapid response. A contract for this work was signed in June of that year, and work on the project has been continuously in progress since that time.

The calorimeter as originally contemplated was essentially completed about a year ago, but the ONR requested that additional equipment be incorporated to permit the independent measurement of the heat exchange which takes place in the respiratory tract.

This additional equipment was completed, and enough tests were made to determine the operating characteristics of the complete assembly. The apparatus was then dismantled, and was shipped to the *Naval Medical Research Institute* at Bethesda, Md., the latter part of December, 1953.

R. G. Huebscher, who had charge of the design and construction of the equipment, will be at the Institute for approximately six months to supervise the assembly of the calorimeter and put it in operation.

INDUSTRIAL ENVIRONMENT

Design of Local Exhaust Ventilation for Hot Industrial Processes (University of Pittsburgh): Any hot process creates a definite pattern of air flow by convection, the magnitude of which is related to the rate of heat release. The objective of a cooperative research project with the University of Pittsburgh is to relate the volume and pattern of this convection flow with the rate of heat release, surface temperature, and physical dimensions of the hot body.

A considerable amount of work was done toward this end; however, the convected stream was so turbulent and unstable that reproducible results were unobtainable.

Recent investigations have been made to directly determine the amount of exhaust ventilation required at various distances above the hot body to capture convected air. Tests have been made for both canopy and lateral hoods of various dimensions, and with heated cylinders of various sizes having various rates of heat release.

Guide Revisions: The TAC on Industrial Environment has prepared a major revision for one chapter and minor revisions for four chapters of THE GUIDE 1954.

INSULATION

Thermal Properties of Building and Insulating Materials: Under the guidance of the TAC on Insulation, members of the Laboratory Staff are making a complete review and analysis of the thermal properties of building and insulating materials. Immediate objectives of this long range program are to revise and improve data in THE GUIDE, and to develop an improved form for their presentation.

Laboratory test data on insulating materials have been requested from a number of insulation manufacturers and trade associations. A number of these have submitted the desired data while others are having new tests made. It is expected that the collection of these data will be completed by early spring, and that the Laboratory Staff and the Committee can start their compilation by that time.

Measurement of Moisture Permeance (Pennsylvania State College): Under a cooperative research contract with the Society, an instrument was developed at The Pennsylvania State College for rapid measurement of water vapor permeance of thin materials of the type commonly used for vapor barriers. The work was completed in 1952, and a paper, Automatic Permeance Testing by the Permeometer, by F. A. Joy and A. W. Sherdon, was presented at the Semi-Annual Meeting in Denver.

ODORS

The installation of the air conditioning equipment for the odor test rooms in the Laboratory was completed during the year. Tests have been made that prove the equipment can independently maintain the desired temperature and relative humidity of each room.

Because of the staff shortage, it has been impossible to get the odor test program under way, but hope persists the right leader for this project will soon be found. A

series of tests to determine effects of temperature and relative humidity on odor perception will be the first experiments in the rooms.

Odor Correlation Conference: On April 23-24, 1953, a Basic Odor Research Correlation Conference was held at the Barbizon-Plaza Hotel in New York, N. Y., under the joint sponsorship of the A.S.H.V.E. and the *New York Academy of Sciences.* Much of the organizational work for this conference was done by the members of the TAC on Odors.

The many papers presented at the conference reported on a great variety of research being done on odors by universities, industrial organizations, and government agencies. The conference was attended by 324 persons vitally interested in some phase of the odor problem.

Proceedings of the conference were published by the *New York Academy of Sciences* (see report in A.S.H.V.E. JOURNAL SECTION, *Heating Piping & Air Conditioning,* June 1953, pp. 129-131).

PANEL HEATING AND COOLING

Group A: To supplement and check the results of earlier tests, two additional plaster panels were constructed and tested at the Laboratory. Special care was taken in plastering one of the panels to obtain unusually good tube embedment; while on the second panel tube embedment was poorer than normal.

Data contained in the paper, Further Studies of the Thermal Characteristics of Plaster Panels, by L. F. Schutrum and C. M. Humphreys, presented at the Semi-Annual Meeting in Denver, indicate that good tube embedment and good contact between tubes and metal lath are both important.

Data on heat flow in a concrete floor panel laid on the ground with tubes at the slab-earth interface, were given in a paper presented by title at the 1953 Semi-Annual Meeting. Data for the paper were determined at the Laboratory by means of an electrolytic type analogue.

Group B: Early in 1952, a series of tests was made in the Environment Laboratory to study heat exchanges in a ceiling panel heated room. Tests were made in an empty room with the four walls and floor maintained at a uniform temperature. The results of this work were presented at the Annual Meeting in January 1953. During the latter part of 1952 and early 1953, a similar series of tests with similar limitations in test conditions was made in a floor panel heated space. The findings of this series were reported in a paper given at the Semi-Annual Meeting in Denver.

These original tests were made under idealized conditions. At the 1954 Annual Meeting, a paper was presented on the modifying effects of non-uniform environment and room furnishings. This paper showed that heat output of a floor or ceiling panel, and the resulting room air temperature, are essentially the same for a non-uniform environment as for a uniform environment having the same average unheated surface temperature. It also indicates that the effect of room furnishings can be neglected without serious error.

This paper also contains information on effects of various combinations of three types of carpets and two types of carpet padding on the heat output of a floor panel.

Tests to determine the effect of room size, and the effect of heating only a part of the floor or ceiling area are now nearing completion, and a paper on this work should be available in the near future.

Another series of tests will soon be started to determine the radiation and convection components of the heat exchanges taking place in a panel heated space.

With the completion of these tests, sufficient information should be at hand to permit drafting a simplified design procedure. A special subcommittee of the TAC on Panel Heating and Cooling has been appointed to work with members of the Laboratory Staff on this next important step.

The next major step in the experimental program under *Group B* will be the study of panel cooling. At its last meeting the Committee outlined the first phase of the panel cooling program, and this should be started soon. Piping circuits for the Environment Laboratory have already been changed to permit heating of the walls for the panel cooling studies.

Group D: Under a Research Fellowship established at the University of Minnesota, field studies were made during the 1951-52 heating season to develop information on the control of floor panel heating systems. The three installations studied included a single-story office building and a factory building, both in Milwaukee, and a residence in Minneapolis. A paper covering the first year's work was presented at the 1953 Annual Meeting.

During the summer of 1952, certain changes were made in the panel systems or their controls, and tests were repeated during the 1952-53 heating season. A paper giving the results of these later tests, Field Studies of Floor Panel Control Systems, Part II, by A. B. Algren, E. F. Snyder, Jr., and R. R. Head, was presented at the Annual Meeting in Houston.

PHYSIOLOGICAL RESEARCH

Physiological Research (University of Illinois Medical School): Under a cooperative research program, supported jointly by the U. S. Public Health Service and the Society, a study was made of the physiological responses of obese, hypothyroid, and hyperthyroid patients to sudden changes in environment. This program has been completed, and a paper giving the results of the study is tentatively scheduled for presentation at the next Semi-Annual Meeting.

PLANT AND ANIMAL HUSBANDRY

Role of Air Conditioning in Plant Growth (California Institute of Technology): A study of the effect of air conditioning on plant growth was made at the Earhart Plant Research Laboratory of the California Institute of Technology, under a cooperative research agreement with the Society. Most of the work was done with tomato plants. The importance of both day and night temperatures was demonstrated, and possible increases of several hundred percent in the yield were indicated. A paper, The Effect of Air Conditioning on Plant Growth, by F. W. Went, was presented at the Annual Meeting in Houston.

SORPTION

Dynamic Characteristics of a Solid Adsorbent (Pennsylvania State University): This project is being carried on under a cooperative agreement with the Society. A literature survey on the subject has been completed and both experimental and analytical studies are under way.

Methods of Testing Sorption Type Dehumidifiers: The TAC on Sorption has continued in its efforts to devise suitable methods of testing sorption type dehumidifiers.

SOUND AND VIBRATION CONTROL

Classification of Research Problems on Sound and Vibration: The TAC on Sound and Vibration Control accepted, with slight modification, the outline of research problems recommended by the Engineering Sub-Committee of *N.A.F.M.* The first problem in the outline relates to instrumentation, and the TAC is now developing a paper on this subject.

WEATHER DATA

A new TAC on Weather Data was appointed during the year and held its organization meeting at the Semi-Annual Meeting in Denver. This Committee plans to confine its activities to the development of comprehensive weather data, and believes that the recommendation of design temperatures is more properly a function of the TA Committees on Heating Load and Cooling Load.

Mr. Dill requested that G. V. Parmelee, Cleveland, Ohio, present some comments on his recent service to the Israeli Government through the United Nations Technical Assistance Board. Mr. Parmelee had accepted a short-term appointment as technical advisor to the Meteorological Service of Israel.

First Vice President Hunter thanked Messrs. Dill and Parmelee for their reports and turned the meeting over to P. B. Gordon who served as chairman of the technical session. Four papers were presented by their authors and discussed.

The session concluded with the report of the tellers of election which was presented by S. W. Beaty, chairman.

REPORT OF TELLERS OF ELECTION

Ballot for Officers	Total
President, L. N. Hunter, Johnstown, Pa.	1777
First Vice President, John E. Haines, Minneapolis, Minn.	1667
Second Vice President, John W. James, Chicago, Ill.	1772
Treasurer, E. R. Queer, State College, Pa.	1770
Members of Council (Three-year Term)	
J. H. Fox, Toronto, Ont.	1759
A. J. Hess, Los Angeles, Calif.	1755
C. H. Pesterfield, East Lansing, Mich.	1751
B. H. Spurlock, Jr., Boulder, Colo.	1745
Committee on Research (Three-year Term)	
W. F. Friend, New York, N. Y.	1757
John Everetts, Jr., Philadelphia, Pa.	1758
C. F. Kayan, New York, N. Y.	1756
H. R. Limbacher, Kalamazoo, Mich.	1755
E. F. Snyder, Jr., Minneapolis, Minn.	1758
TOTAL BALLOTS RECEIVED	1955
TOTAL LEGAL BALLOTS	1779
INVALID BALLOTS	176
TOTAL WRITE-INS	14

Respectfully submitted,

S. W. BEATY, *Chairman*
A. B. ULLRICH, JR.
R. E. HAUBOLD

Following receipt of the report the meeting was adjourned at 12:30 p.m.

THIRD TECHNICAL SESSION, WEDNESDAY, JANUARY 27TH, 9:30 A.M.

The third technical session was called to order by Second Vice Pres. J. E. Haines who introduced Prof. B. H. Jennings, chairman of the technical session.

Three technical papers were presented by their authors and discussed. Professor Jennings thanked the authors and the discussors for their contributions.

Mr. Haines resumed the chair and adjourned the session at 12:15 p.m.

FOURTH TECHNICAL SESSION, WEDNESDAY, JANUARY 27TH, 2:00 P.M.

The fourth and last session of the meeting was called to order by Pres. Reg. F. Taylor at 2:00 p.m. W. A. Grant, Syracuse, N. Y., was chairman of the technical session and introduced the authors of the three technical papers.

At the conclusion of the discussion of the three papers, President Taylor resumed the chair and called for the report of the Resolutions Committee. The report was presented by B. L. Evans, chairman, and was adopted unanimously.

RESOLUTIONS

WHEREAS, No set of resolutions appears adequate to tell of the virtues of this great state which the speakers have so modestly refrained from extolling, and

WHEREAS, One large ship in a small pond (we understand the reason it is small is because they need so much cooling water for their air conditioning) one small star on a flag and one small monument on the plains appear entirely in keeping with the modest statements of Texans, and

WHEREAS, The 60th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held in the most air conditioned metropolis in the world is now drawing to a close, and

WHEREAS, The aims of the Society have been furthered through the presentation and discussion of outstanding technical papers under the capable direction of our officers and staff, and

WHEREAS, The enjoyment of Members and Guests has been greatly enhanced due to carefully planned social, cultural and recreational events of a delightful nature, and

WHEREAS, All these combined with a record attendance and the enthusiasm of the membership have made this an outstanding meeting; therefore,

BE IT RESOLVED, That we express our sincere gratitude and thanks:

TO D. M. Mills, general chairman of the Committee on Arrangements,

TO R. J. Salinger, vice chairman, and

TO all committee chairmen and members of their committees who planned and conducted this most pleasant meeting.

TO Pres. E. G. Floeter, Jr., and the officers and members of the South Texas Chapter who contributed so much to the success of this meeting,

TO Honorary Chairman C. Rollins Gardner for his willingness to be banquet toastmaster,

TO the other Texas Chapters for their cooperation and support,

TO Mayor Roy Hofheinz and the City of Houston for their gracious welcome and hospitality, and

TO Morris Frank for his modest and humorous toastmastership at the Welcome Luncheon.

TO the daily and technical press for the support and publicity given this meeting,

TO the Chamber of Commerce and the Hotel Management for housing and caring for our needs, and the excellent service provided by the personnel of The Rice,

TO the authors and discussors of the technical papers,

TO the Ladies of Texas who so graciously assisted in the hospitality and made the stay of the visiting ladies so enjoyable,

TO Dr. W. W. Kemmerer for his courtesy in preparing his talk on An Educator Looks at Air Conditioning for our banquet,

TO Walter L. Fleisher whose contribution to our Society's aims has here been recognized by the award of the F. Paul Anderson Medal, and

TO all others who have contributed to the success of this meeting by their work and attendance.

Respectfully submitted,

The Resolutions Committee

B. L. EVANS, *Chairman*, St. Louis, Mo.
F. H. FAUST, Bloomfield, N. J.
C. B. GAMBLE, New Orleans, La.

(The installation of the officers was conducted at the Banquet on Wednesday evening.) The technical session was adjourned at 4:30 p.m.

PROGRAM—60th ANNUAL MEETING

The Rice Hotel, Houston, Texas

January 25-27, 1954

Saturday—January 23

10:00 a.m. Executive Committee Meeting *(Western Room)*

2:00 p.m. Finance Committee Meeting *(Western Room)*

2:00 p.m. Ways and Means Committee Meeting *(Room E)*

Sunday—January 24

10:00 a.m. REGISTRATION *(Mezzanine Floor—Lounge)*

10:00 a.m. Council Meeting *(French Room)*

10:00 a.m. Committee on Research *(Hunt Room)*

2:00 p.m. Motor tour through beautiful residential section of Houston

4:00 to
6:00 p.m. Rendezvous for coffee Texas style *(Sam Houston Room)*
Hostesses: Mrs. Reg. F. Taylor, Assisted by:
Mmes. E. G. Floeter, Jr., J. W. Holland, J. C. Lewis, F. M. Neil, O. G. Rivoire, A. B. Ullrich, Jr.

7:00 p.m. Dinner, Floor Show, and Dancing at the Shamrock Hotel

Monday—January 25

9:00 a.m. REGISTRATION *(Mezzanine Floor—Lounge)*

9:00 a.m. Hospitality Room *(Sam Houston Room) Hosts*—North Texas Chapter, Dallas

9:30 a.m. FIRST TECHNICAL SESSION *(Ballroom)*
Call to order by Pres. Reg. F. Taylor
Greetings by D. M. Mills, General Chairman
Reports of Officers and Council
Amendments to By-Laws
L. T. Mart, *Chairman*
Prediction of Cooling Tower Performance, by W. W. Smith, College Station, Tex., presented by Mr. Smith
Automobile Air Conditioning—Progress and Problems, by P. J. Kent, Detroit Mich., presented by Mr. Kent
Pressure Loss of Air Flowing Through 45-Degree Wooden Louvers, by Phil R. Cobb, College Station, Tex., presented by Prof. W. D. Scoates, College Station

12:15 p.m. WELCOME LUNCHEON *(Ballroom)* E. G. Floeter, Jr., President, South Texas Chapter, presiding
Toastmaster: Morris Frank of the *Houston Chronicle*
Welcome to Houston: Mayor Roy Hofheinz

2:00 p.m. Chapters Conference Committee Meeting *(Jade Room)*

2:00 p.m. TAC on Heat Flow Through Glass, R. A. Miller, *Chairman (French Room)*

2:00 p.m. TAC on Heat Pump, E. P. Palmatier, *Chairman (Room E)*

2:00 p.m. TAC on Plant and Animal Husbandry, A. J. Hess, *Chairman (Western Room)*

2:30 p.m. Boat and bus trip to San Jacinto Monument—See Ship Channel, Industrial District; Visit Battleground and Battleship Texas

3:00 p.m. TAC on Weather Data, John Everetts, *Chairman (Room 336)*

6:00 p.m. Dinner at San Jacinto Inn (served family style). Return to Rice Hotel by bus at 8:30 p.m.

Tuesday—January 26

9:00 a.m. REGISTRATION *(Mezzanine Floor—Lounge)*

9:00 a.m. Hospitality Room *(Sam Houston Room) Hosts*—Southwest Texas Chapter, San Antonio

9:30 a.m. SECOND TECHNICAL SESSION *(Ballroom)*

Call to order by First Vice Pres. L. N. Hunter

Report of Committee on Research—R. S. Dill, *Chairman*

P. B. Gordon, *Chairman*

Circuit Analysis Applied to Load Estimating, by H. B. Nottage, Encino, Calif., and G. V. Parmelee, Cleveland, Ohio, presented by Mr. Parmelee

Heat Flow Characteristics of Hot Water Floor Panels, by E. L. Sartain and W. S. Harris, Urbana, Ill., presented by Mr. Sartain

Effects of Non-Uniformity and Furnishings on Panel Heating Performance, by L. F. Schutrum and C. M. Humphreys, Cleveland, Ohio, presented by Mr. Schutrum

Field Studies of Floor Panel Control Systems, Part II, by A. B. Algren, E. F. Snyder, Jr., and R. R. Head, Minneapolis, Minn., presented by Mr. Snyder

Report of Tellers of Election

12:30 p.m. Ladies Luncheon and Fashion Show *(Empire Room)*
Presented by Battlestein

1:00 p.m. Long Range Planning Committee, J. E. Haines, *Chairman (Western Room)*

1:30 p.m. Nominating Committee Meeting *(French Room)*

2:00 p.m. TAC on Evaporative Cooling, A. J. Hess, *Chairman (Room 336)*

2:00 p.m. Joint Committee for Revision of Code of Minimum Requirements for Comfort Air Conditioning, W. L. Fleisher, *Chairman (Room E)*

2:00 p.m. Inspection trips to Public, Industrial and Commercial Buildings

7:00 p.m. Authentic Mexican Dinner *(Ballroom)*

7:00 p.m. Past Presidents' Dinner *(Room E)*

8:00 p.m. Informal Entertainment—Movies, TV shows—Cards, Music and Dancing *(Mezzanine Floor)*

Wednesday—January 27

9:00 a.m. REGISTRATION *(Mezzanine Floor—Lounge)*

9:00 a.m. Hospitality Room *(Sam Houston Room) Hosts*—West Texas Chapter, Lubbock, Texas

9:30 a.m. THIRD TECHNICAL SESSION *(Ballroom)*

Call to order by Second Vice Pres. John E. Haines

B. H. Jennings, *Chairman*

Costs of Operating the Heat Pump in the Equitable Building, by J. B. Kroeker and R. C. Chewning, Portland, Ore., presented by Mr. Kroeker

Availability and Utilization of Solar Energy, by R. C. Jordan and J. L. Threlkeld, Minneapolis, Minn., presented by Drs. Jordan and Threlkeld

Part 1. Solar Energy Availability for Heating in the United States

Part 2. Utilization of Solar Energy for House Heating

Part 3. Design and Economics of Solar Energy Heat Pump Systems

Effects of Air Conditioning on Plant Growth, by F. W. Went, Pasadena, Calif., presented by Dr. Went

12:00 noon TAC on Sorption, G. L. Simpson, *Chairman (Western Room)*

1:30 p.m. Ladies Theater Party *(Playhouse Theater)* Buses leave Rice Hotel at 1:30 p.m. Curtain time 2:00 p.m.

2:00 p.m. FOURTH TECHNICAL SESSION *(Ballroom)*

Call to Order by Pres. Reg. F. Taylor

W. A. Grant, *Chairman*

Room Air Distribution Research for Year 'Round Air Conditioning —Part II, Supply Outlets at Three Floor Locations, by H. E. Straub, Urbana, Ill., and S. F. Gilman, Syracuse, N. Y., presented by Mr. Straub

Cooling a Small Residence Using a Perimeter-Loop Duct System, by D. R. Bahnfleth, C. F. Chen and H. T. Gilkey, Urbana, Ill., presented by Mr. Bahnfleth

A Two-Sphere Radiometer, by D. J. Sutton and P. E. McNall, Jr., Minneapolis, Minn., presented by Mr. Sutton

Resolutions

7:00 p.m. ANNUAL BANQUET *(Ballroom)*

Call to order

Toastmaster: C. Rollins Gardner

Invocation

Presentation of F. Paul Anderson Medal to Walter L. Fleisher, by Reg. F. Taylor

Installation of Officers

Presentation of Past President's Emblem to Reg. F. Taylor

Speaker: Dr. W. W. Kemmerer, Executive Director of the Houston World's Fair

Subject: An Educator Looks at Air Conditioning

Dancing

Thursday—January 28

9:30 a.m. Organization Meeting of Council *(French Room)*

COMMITTEE ON ARRANGEMENTS

D. M. MILLS, *General Chairman*

R. J. SALINGER, *Vice-Chairman*

Honorary Chairmen

C. R. GARDNER, Fort Worth

W. E. LONG, Austin

Banquet—A. B. Ullrich, Jr., *Chairman;* J. A. Wheeler, D. M. Robinson, A. M. Chase, Jr., J. N. Powell.

Entertainment—F. M. Neil, *Chairman;* N. L. Brown, D. S. Freese, M. E. Linn, H. E. Hope, Jr., O. G. Rivoire, Wesley Brown, D. C. McNeil, Reese Brentzel, H. R. Safford.

Finance—A. J. Natkin, *Chairman;* Willard Ammons, J. A. Poole.

Ladies—C. L. Fleming, *Chairman;* C. C. Quin; Mmes. A. F. Barnes, C. V. Chenault, D. S. Cooper, B. P. Fisher, C. L. Fleming, E. G. Floeter, J. C. Fontana, G. D. Hines, J. W. Holland, J. C. Lewis, D. M. Mills, I. A. Naman, E. H. McLane, F. M. Neil, W. N. Pauley, Dana Price, C. C. Quin, O. G. Rivoire, R. J. Salinger, W. A. Sharp, R. F. Taylor, A. B. Ullrich.

Publicity—B. P. Fisher, *Chairman;* Wayne Holland, W. J. Way, II, B. J. Beaird, C. W. Warren.

Reception—E. G. Floeter, *Chairman;* Sidney Bromberg, H. O. Williams, J. C. Lewis, M. B. Peiser, G. T. Kerbow.

Sessions—I. A. Naman, *Chairman;* J. B. Buckley, W. L. Lashley, Jr., Bruce Appling, J. C. Fontana, Dallas Sullivan, G. D. Hines, Norman Rosenberg, S. F. Atlas.

Special Events—H. W. Broadwell, *Chairman;* B. P. Fisher, D. S. Cooper, F. R. Young, J. H. Stephenson, M. B. Peiser, H. G. McKee.

Transportation—A. F. Barnes, *Chairman;* A. E. Hess, C. V. Chenault, A. B. Banowsky, R. S. Sandifer, D. W. O'Brient.

No. 1494

PREDICTION OF COOLING TOWER PERFORMANCE

By Wesley W. Smith*, College Station, Tex.

THE NEED for closer evaluation and prediction of mechanical draft cooling tower performance is becoming more noticeable as more and more installations are being put into operation. Some of these installations cost hundreds of thousands of dollars. While many manufacturers have the experience and knowledge needed to build equipment that meets specified requirements, the fact remains that there are many towers in operation today that have never been able to meet their performance specifications.

The many variables (some controlled, others uncontrollable in actual operation) make the determination of the performance, even at a given time, a difficult task. The prediction of operations under other than tested conditions becomes still more difficult, and might be practically impossible with any degree of assurance were it not for the evolution and development work by Robinson[1], McAdams[2], Merkel[3], Nottage[4], Lichenstein[5], and others whose academic considerations of a correlation method were used as the basis for the approximate method described in this paper.

Cooling Tower Approach Curves

Fortunately, the most important external factors bearing on the performance of a counterflow cooling tower have been grouped so that their effects are determined from solution of the equation[6]:

$$\frac{K\,a\,V}{L} = \int_{t_2}^{t_1} \frac{dt}{h'' - h_a} \quad \ldots\ldots\ldots\ldots \quad (1)$$

in which

K = water evaporated, pounds per (square foot of exposed surface) (hour) (unit vapor content differential).
a = exposed surface area, square feet per cubic foot of active tower volume.
V = active tower volume, cubic feet per square foot of tower ground area.
L = water flowing through the tower pounds per (hour) (square foot of ground area).
G = dry air flowing through the tower pounds per (hour) (square foot of tower ground area).

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* Associate Research Engineer, Texas Engineering Experiment Station, Texas A. & M. College System.

[1] Exponent numerals refer to Bibliography.

Presented at the 60th Annual Meeting of The American Society of Heating and Ventilating Engineers, Houston, Tex., January 1954.

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COOLING TOWER APPROACH CURVES

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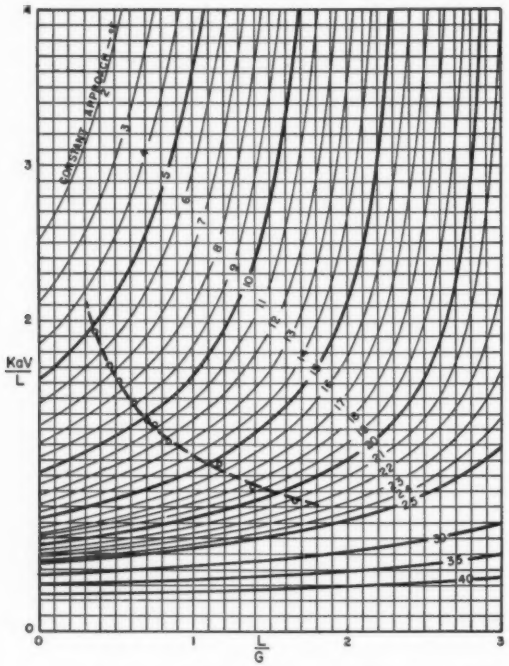


FIG. 1. COOLING TOWER APPROACH CURVES (70 F WET BULB WITH 20 F RANGE AT SEA LEVEL BAROMETRIC PRESSURE)

t_1 = temperature of the water entering the tower, Fahrenheit.
t_2 = temperature of the water off of tower, Fahrenheit.
t = temperature of the water in the tower, Fahrenheit.
h'' = enthalpy of saturated air at temperature t, Btu per pound of dry air.
h_a = enthalpy of air stream in tower at position where water temperature is t, Btu per pound of dry air.

Fig. 1, constructed from this equation, represents the currently usable values of KaV/L and L/G for sea level operation at a wet bulb temperature of 70 F and a range of 20 deg. The circled points were obtained from tests of an 8- by 8-ft experimental tower, while the dotted line represents the probable tower characteristic curve. Fig. 2 contains the same information as Fig. 1 plotted on logarithmic scales for KaV/L and L/G. The tower characteristic curve appears in Fig. 2 as a straight line whose equation can be written:

$$KaV/L = 1.14\ (L/G)^{-0.54} \quad \ldots\ldots\ldots\ldots \quad (2)$$

from which

$$KaV = 1.14\ L^{0.46}\ G^{0.54} \quad \ldots\ldots\ldots\ldots \quad (3)$$

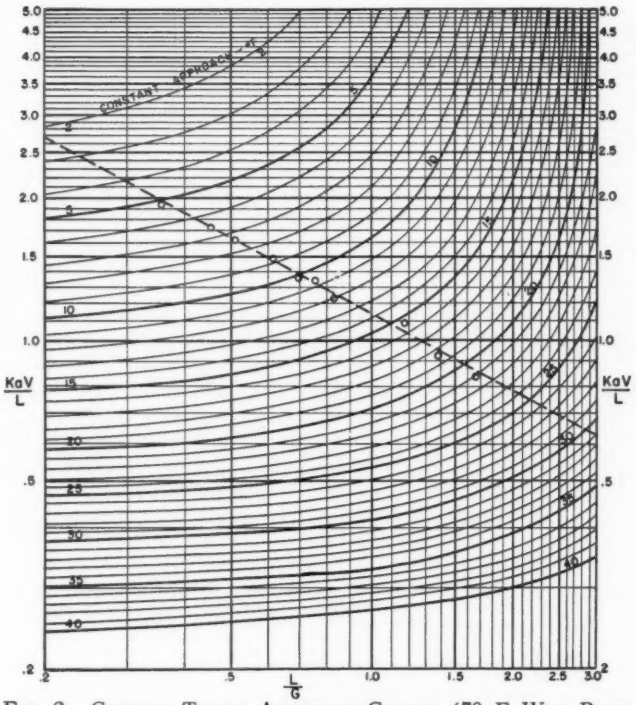


FIG. 2. COOLING TOWER APPROACH CURVES (70 F WET BULB WITH 20 F RANGE AT SEA LEVEL BAROMETRIC PRESSURE)

As no pattern has yet been discovered to explain the value of the multiplier or the exponents, a tower curve must be established by actual test. However, the majority of actual towers appear to have characteristic curves that are very nearly straight on logarithmic paper and therefore two points will suffice to establish the line for a reasonably accurate performance criterion.

When a tower curve has been established, prediction of tower performance becomes a matter of superimposing this curve over the proper set of approach curves. Unfortunately approach curves are not at present available for all the conditions encountered, and probably will not be obtained except by some co-operative effort of those who require them. If curves were constructed for each 2 deg interval of range from 8 F to 50 F, and for wet bulb temperatures in 2 deg intervals from 32 F to 90 F the result would be 660 drawings of approximately 40 curves each. If this construction were completed now for 1000-ft intervals of elevation up to and including 10,000 ft, the number of drawings would become 7260.

Figs. 2, 3, 4, and 5 are presented as an approximate solution to the problem covering 55 F to 85 F wet bulb temperature, 10 to 30 deg range, sea level to

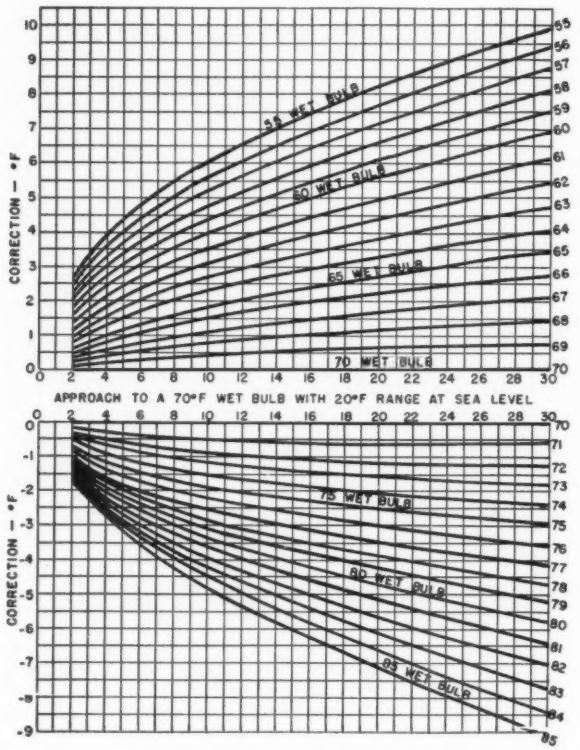


FIG. 3. WET-BULB TEMPERATURE CORRECTION (TO BE APPLIED TO THE APPROACH TO A 70 F WET BULB WITH 20 F RANGE AT SEA LEVEL BAROMETRIC PRESSURE)

8000 ft elevation, and L/G ratios between 0.5 and 2.0. All prediction calculations are based on sea level performance with 70 F wet bulb temperature and 20 deg range. Proper corrections for wet bulb temperature, range, and elevation are applied to the approach under these conditions to compensate for different wet bulb temperatures, ranges, and elevations. The following example will serve to clarify the method:

Example 1: The experimental tower with the characteristic curve of Fig. 2 is to operate at an L/G of 1.20 at 7000 ft elevation, with a wet bulb of 62 F, and with a 25 deg range. Determine the inlet and outlet water temperatures.

Solution:

Item 1. Approach obtained from Fig. 2 for L/G of $1.2 = 16.1$

Item 2. Correction to approach of *Item 1* from Fig. 3 and 62 F $= +4.0$

Item 3. Item 1 plus (algebraically) *Item 2* $= 20.1$

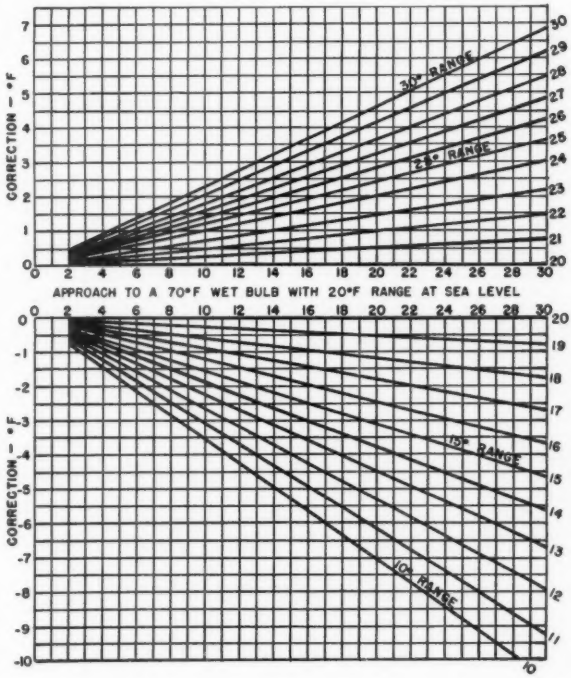


FIG. 4. RANGE CORRECTION (TO BE APPLIED TO THE APPROACH TO A 70 F WET BULB WITH 20 F RANGE AT SEA LEVEL BAROMETRIC PRESSURE)

Item 4. Correction to approach of *Item 1* from Fig. 4 and 25 deg $= +2.0$

Item 5. Item 3 plus (algebraically) *Item 4* $= 22.1$

Item 6. Correction to approach of *Item 1* from Fig. 5 and 7000 ft $= -4.1$

Item 7. Item 5 $-$ *Item 6* $= 18.0$

Item 7 is the required approach $= 18$ deg.

The temperature of water leaving tower = sum of wet bulb temperature, 62 F, and 18 deg approach = 80 F.

The temperature of water entering tower = temperature of water leaving tower + range = 80 + 25 = 105 F.

AIR FLOW

All preceding calculations have been based on the assumption that the same L/G could be maintained even for significant changes in air density resulting from large changes of elevation. For the same water loading, L/G can be maintained constant at altitudes, but this necessitates a change in fan char-

acteristics and an increase in horsepower. If the altered combination has the same efficiency as the sea level installation, then it can be shown that under certain idealized conditions for constant G:

$$P_x = P_o (\rho_o/\rho_x)^2 = P_o (V_x/V_o)^2 \quad \ldots\ldots\ldots\ldots \quad (4)$$

or for same temperature and relative humidity:

$$P_x = P_o (B_o/B_x)^2 \quad \ldots\ldots\ldots\ldots \quad (5)$$

where

P_x = horsepower at altitude x.
P_o = horsepower at sea level.
ρ_o = air density at sea level.
ρ_x = air density at altitude x.
V_x = air specific volume at altitude x.
V_o = air specific volume at sea level.
B_o = barometric pressure at sea level.
B_x = barometric pressure at altitude x.

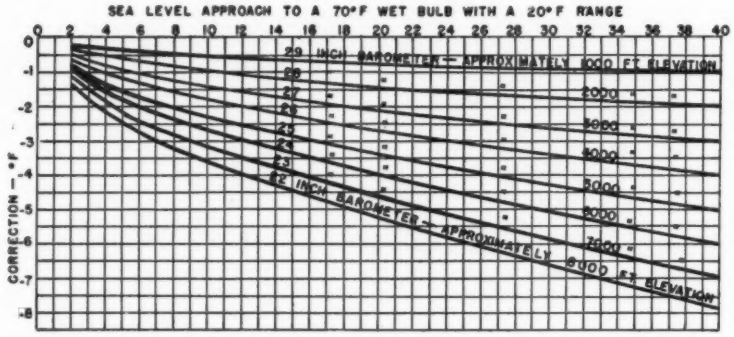


FIG. 5. ELEVATION CORRECTION (TO BE APPLIED TO THE APPROACH TO A 70 F WET BULB WITH 20 F RANGE AT SEA LEVEL TO FIND THE APPROACH FOR OTHER BAROMETRIC PRESSURES)

The horsepower for *Example 1* would be:

$$P_{7000} = P_o (30/23)^2 = 1.7\ P_o$$

This 70 percent increase in horsepower, in addition to being a considerable first cost item, would also mean a considerable increase in operating expense. Therefore, the advisability of using a smaller quantity of air, G, should be investigated.

The expected relation for horsepower as a function of G and ρ is:

$$P_x = P_o \left(\frac{G_x}{G_o}\right)^3 \left(\frac{\rho_o}{\rho_x}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (6)$$

For constant temperature and humidity this would become:

$$P_x = P_o \left(\frac{G_x}{G_o}\right)^3 \left(\frac{B_o}{B_x}\right)^2 \quad \ldots\ldots\ldots\ldots \quad (7)$$

If in *Example 1* the horsepower were kept constant, (the fan would, in all probability, require alteration), the air flow at 7000 ft altitude would become:

$$G_{7000} = G_o\,(23/30)^{2/3} = 0.84G_o$$

With a constant water loading, this would result in a change from a value of L/G of 1.20 to a value of 1.43. The example tower would have an approach of 20.6 deg. It should be noted that this approach is better than the 22.1 deg approach of the same tower with the same horsepower operated at sea level.

The horsepower required to maintain the same volumetric rate of flow at 7000 ft, as at sea level, would be:

$$P_{7000} = P_o\,(\rho_x/\rho_o) = 0.77\,P_o$$

The resulting G and L/G for constant water loading would be, respectively, $0.77G_0$ and 1.57. The 7000 ft tower would have an approach of 21.9 deg which is practically the same as the 22.1 deg of the sea level tower. Note that it is also probable that no change in fan characteristics would be necessary for this operating condition.

CONCLUSIONS

The tower evaluation methods described involving the expression,

$$\frac{K\,a\,V}{L} = \int_{t_2}^{t_1} \frac{dt}{h'' - h_a}$$

has been found to give reasonably good results although influenced by factors which at present either are not recognized or whose influences are not adequately evaluated. Construction and wide distribution of a complete set of graphs similar to Fig. 2 would naturally lead to better understanding of cooling tower phenomena; improve the customer-manufacturer relationship; and tend to minimize unethical practices.

The approximate method of tower evaluation described herein and involving the use of Figs. 2, 3, 4, and 5, although admittedly rough, nevertheless is probably much better than the methods usually used by the cooling tower operator. The altitude analysis is idealized but based on sound fundamentals and therefore is believed to be generally correct. Until experimental verifications of altitude effects are obtained, the foregoing can serve as a guide to cooling tower performance at elevations.

The improvement in tower performance with increase in altitude is considerable if the same weight of air is handled. However, as this involves a significant increase in fan horsepower for large altitude increases, a reduction in the value of G may be advisable with either a similar reduction in L to maintain a constant L/G or with an unchanged L resulting in an increase in L/G. *Example 1* discussed in this paper is interesting especially in illustrating that virtually the same performance was expected for a 7000 ft elevation as for sea level but with about one-quarter decrease in fan horsepower.

ACKNOWLEDGMENT

The author wishes to acknowledge the assistance of Lilie-Hoffmann Cooling Towers, Inc. in supplying equipment and also practical advice on cooling tower construction and operation.

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3. Verdunstungskuhlung, by F. Merkel (*VDI Forschungsarbeiten,* No. 275, 1925).

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5. Performance and Selection of Mechanical Draft Cooling Towers, by Joseph Lichenstein (*A.S.M.E. Transactions,* Vol. 65, 1943, p. 779).

6. HEATING VENTILATING AIR CONDITIONING GUIDE, 1953, p. 776 (published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 62 Worth St., New York 13, N. Y.).

DISCUSSION

H. E. DEGLER, Kansas City, Mo. (WRITTEN): This paper is another contribution to the available A.S.H.V.E. reference material on cooling tower performance. Of particular interest are the groups of curves (shown in Figs. 2, 3 and 4) which were prepared from test data and correlated calculations. However, photos and details of construction and operation of the experimental tower would have been of equal value and helpful to show the basic type of tower filling, the size and shape of the splash bars, their spacing horizontally and vertically, whether the water was pressure sprayed onto the top layer of fill, and correlated operational details.

The writer made a number of exploratory calculations by applying this information to two models of counterflow towers, using 0.75 to 2.5 liquid-gas ratios, 60 to 80 F wet-bulb temperatures, and 10 to 30 deg cooling range with 6 to 22 deg approach. The predicted performance with the use of Smith's graphs closely checked the actual capabilities of these towers.

In general, cooling tower performance improves with altitude. For a given tower, the same motor and fan horsepower would be used regardless of altitude, hence a greater volume of air is circulated than at sea level; this effect is attained by increasing the pitch of the fan blades. In the last sentence of his conclusions the author states that there would be a *quarter decrease in fan horsepower at 7000 ft elevation.* This is probably based upon the decrease in density of the air of which a greater volume is circulated. However, the enthalpy per pound of air increases with altitude. These compensating properties of air for altitude installations are reflected by the author in the section under Air Flow.

It should be mentioned that at elevations above 3300 ft, motor design modifications would be necessary to keep the motor within safe operating temperatures.

The author mentioned that large numbers of drawings are required for a more complete performance picture. We use hundreds of curves and several thousand tabulated figures for the basic rating and selection data for each cooling tower design. Some established manufacturers may have ten or more designs with different flow patterns and splash-bar arrangements for a wide variety of temperatures and capacities, each requiring a separate mathematical and graphical analysis including basic experimental correlation. Each American cooling tower manufacturer has a semi-confidential method of sizing towers based upon research and actual performance correlated with definite requirements.

Because of the many variables in cooling tower calculations and performance, it is difficult to provide simple, handbook equations, tables and graphs, whereby a non-

specialist could readily select the type and size of tower for any one of the many possible combinations of desired water quantities, temperatures and applications.

T. J. CONNOLLY* and J. C. HARPER,* Los Angeles, Calif. (WRITTEN): The curves prepared by the author would be of use to plant operators in evaluating effects of changes in operating conditions and to engineers for preliminary design work. The curves presented are plots of solutions to the usual cooling tower equations, and the results should be considered as a convenient handbook type of information rather than as a fundamental contribution to cooling tower theory. It is difficult to understand how the compilation of a complete set of such curves would lead to a better understanding of cooling tower phenomena.

The reviewers would like to ask the author if he is familiar with the set of charts prepared by the Foster Wheeler Corp. In this collection are approximately 450 charts identical to the author's Fig. 1, covering different wet-bulb temperatures and cooling ranges, but not including the effect of barometric pressure.

The value of this paper would be increased if the author were to give an estimate of the magnitude of the errors incurred by the use of Figs. 3, 4, and 5. These corrections must be affected to some extent by the value of L/G in addition to other variables included. The derivation of Equations 4 and 6 should also be given, together with any postulates made.

Experimental results with gas-liquid contacting equipment in general indicate that transfer coefficients vary approximately as the 0.8 power of G. The reviewers would like to ask if the result indicated in Equation 3 is typical, and what range of exponents of G might be expected.

H. B. NOTTAGE, Encino, Calif. (WRITTEN): This paper is disappointing in its lack of useful and extensive experimental performance data which really would be needed to properly validate the performance prediction method. For instance, may we have the complete construction details and original test data for the tower(s) whose performance is given by Fig. 1?

Variations of this same method have recently been expounded by other authors, for example Baker and Mart† and Mickley‡.

W. M. WALLACE, II, Durham, N. C.: I want to compliment Mr. Smith for the introduction of some data which will be of use to engineers. We have received from several companies so-called confidential data on cooling tower selection. It is practically impossible to correlate one manufacturer's data with another and it poses for the engineer a very difficult problem of making comparisons. As Mr. Degler pointed out, his firm has hundreds of curves. Reference was made to other manufacturers who have similar curves. It seems that it would be quite possible to develop the type of curve shown in this paper and produce curves which would apply to any manufacturer's equipment by using necessary correction factors.

It is my hope that out of this paper will come more correlation of the data from different manufacturers so that so much difficulty will not be experienced in comparing the performance of cooling towers of different manufacturers.

L. T. MART, Kansas City, Mo.: The *Cooling Tower Institute* of which I am president has been in existence for about four years. One of its objectives is the preparation of data and charts which will enable the buyer to test the cooling towers and determine by reasonably accurate measurements the deficiency involved. I believe we are not far from having a practical commercial answer to that problem.

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* Department of Engineering, University of California.

† Analyzed Cooling Tower Performance by This Method, by D. R. Baker and L. T. Mart (*The Petroleum Refiner*, October 1952, p. 97).

‡ Design of Forced-Draft Air-Conditioning Equipment, by H. S. Mickley (*Chemical Engineering Progress*, Vol. 45, December 1949, p. 739).

In this paper, as in others, characteristics of tower performance are shown as well as the testing done and results obtained. However, the charts now available show insufficiencies but do not indicate their degree. This is one of the problems to which I think we will soon have a solution.

AUTHOR'S CLOSURE: Mr. Wallace's comments and Mr. Degler's complimentary remarks and verification of the prediction method are greatly appreciated.

The tower which provided the test results was an $8 \times 8 \times 20$-ft induced draft, counterflow type with redwood packing and spray distribution. Further details cannot be disclosed at present because of proprietary sponsorship of the work. It should perhaps be re-emphasized that the test data from the test tower were incidental and were used only for illustration. All conventional towers tested by the author have been in essential agreement with the equation.

$$KaV/L = S(L/G)^{R}$$

The value of S varied through a wide range depending on the service for which the tower was intended; the value of R, however, varied through the narrow range of 0.50 to 0.65.

Mssrs. Connolly and Harper have perceived the principal uses of the performance prediction method under discussion, but evidently do not share the author's belief that understanding in the engineering sense must include functional familiarity. The charts noted by them would be very valuable if generally available to the engineering profession.

It is difficult, if not impossible, to give a general answer to the question concerning the degree of approximation introduced by the use of Figs. 3, 4, and 5. Equation 1 is in itself an approximation depending on many factors. The L/G ratio does have a bearing on the results. This was taken into account in the paper when the results were restricted to L/G ratios between 0.5 and 2.0.

Equations 4 and 6 are readily obtained from the fan laws providing one thinks in terms of a fan of 1 sq ft cross-sectional area.

Fan law No. 3*, can then be written as: horsepower is proportional to the volumetric flow cubed and to the density to the first power

$$P \propto Q^3 \rho$$

And the weight is proportional to the linear power of both the volume and the density

$$G \propto Q\rho$$

$$\therefore Q \propto \frac{G}{\rho}$$

and

$$P \propto \left(\frac{G}{\rho}\right)^3 \rho = G^3/\rho^2$$

From which:

$$P_x = P_o \left(\frac{G_x}{G_o}\right)^3 \left(\frac{\rho_o}{\rho_x}\right)^2$$

which is Equation 6.

Equation 4 is a special case of Equation 6 with G constant.

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* *Fan Engineering*, Buffalo Forge Co., Buffalo, N. Y. (Fifth Edition, p. 242).

1495

No. 1495

AUTOMOBILE AIR CONDITIONING — PROGRESS AND PROBLEMS

By P. J. KENT*, DETROIT, MICH.

BETWEEN THE YEARS of 1939 and 1953 approximately 10,500 cars have been air conditioned, but during this period none of the automobile companies had produced their own air conditioning system. However, during the past year several of the major companies have built systems of their own design for installation in their automobiles.

It can be fairly stated that none of the companies were able to supply the demand for air conditioning on their cars during the spring, summer and early fall months of 1953. Suppliers outside the automobile industry have also enjoyed an excellent market during the past year.

Additional automobile companies are planning to offer air conditioning in 1954, and those companies which had such systems in 1953 are planning greatly increased volume of production, so the prediction made in March that 10 percent of all new cars would be equipped with air conditioning within five to ten years would appear to be a pretty fair estimate. It has been definitely established that a large percentage of people living in the warmer sections of the country will pay the additional cost required to obtain air conditioning on the medium and high-priced cars.

TOOLING FOR AIR CONDITIONING

A year ago the problem of tooling and tool amortization was approached with considerable caution by production departments. Without knowing what the market would be or how stable the design might prove, there was considerable reluctance to spend money for expensive dies, molds and fixtures. As a result, many parts were partially fabricated by hand or machined from sand castings. In some cases, easily formed but expensive plastic parts were substituted for metal stampings with the net result that tooling costs were relatively low but piece prices high.

With every indication that the market will grow quite rapidly, a gradual change to mass production methods can be expected. However, this will not necessarily mean immediate price reduction, because while piece prices will drop, tool costs will increase. Only when production volume is achieved will the overall cost come down.

Air conditioning of offices and commercial establishments has had a very rapid growth in the last five or six years, and within the last two or three years the same trend in home air conditioning has been observed. If the progress of automobile air conditioning follows the same pattern, and there is every reason to believe it will, then appreciable price reductions based on savings resulting from mass production methods should be soon forthcoming.

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* Executive Engineer, Electrical Section, Chrysler Corp.

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ASSEMBLY METHODS

In the plant the initial planning called for completing the car assembly on the regular assembly line and then delivering it to a special department where the modifications and assembly of air conditioning equipment were made. However, it was apparent that the major air conditioning units could be assembled in the car on the regular automobile assembly line leaving only the tubing assembly, charging and testing to be done in a special department.

ENGINEERING THE SYSTEM

Operating experience with pre-war automobile air conditioning and at least one of the post-war systems, proved that a system should have a large amount of reserve capacity over that required for most normal driving conditions. An automobile installation is different from a stationary one because it cannot be left operating when the car is not in use or is unoccupied by passengers.

Large refrigeration capacity offers several advantages:

1. When a car has been parked in the sun, large quantities of heat are stored within, and refrigeration capacity must be sized sufficiently large to lower the temperature quickly. Many automobile trips are of short duration so quick cooling is necessary if the system is to have value.

2. There are times when the engine is idled or operated at very low speed over a considerable period. Therefore, it is important that the system have a reasonably good output during low speed operation.

3. Experience over the past year has demonstrated that owners will idle the motor of an air conditioned car much longer than one which is not air conditioned, and this has introduced additional problems in connection with the compressor drive and engine cooling.

4. Good general passenger compartment cooling can be obtained without resorting to high velocity air movement. By using slow velocity air throughout the car, local drafts, which cool some parts of the passenger's body more than others, can be avoided. The system which cools by localized drafts has been compared to that in an airplane, but most people prefer the cooling as found in a modern Pullman car.

5. Quietness of operation can be attained. One of the reasons for air conditioning in an automobile is the ability to drive with the windows closed, shutting out road noise and dust. It does not seem logical to merely substitute the noise of numerous high speed air jets in the car for the outside road noise and wind noise experienced when driving with windows open in cars which are not air conditioned.

INTRODUCTION OF OUTSIDE AIR

Most of the automobile air conditioning systems are now using from 15 to 25 percent outside air, and from 75 to 85 percent recirculated. This again calls for a higher capacity refrigeration system. The use of outside air has two advantages, (1) to get rid of smoke and odors which tend to accumulate in a car using recirculated air only and, (2) the introduction of outside air builds up a slight pressure in the passenger compartment which prevents the infiltration of dust and warm outside air through the numerous small leaks around doors, windows, etc. Experience has proved the use of 25 percent outside air to be successful.

CONDENSATION

In one unit the flush louvered grille on the package shelf, through which the chilled air was delivered to the passenger compartment was changed to an out-

let with louvers about 3 in. high to keep the cold air from striking the rear window and causing condensation or fog to form on the outside. This outlet also directs the cold air over the head of passengers sitting in the center of the rear seat and generally improves distribution through the car.

In those systems which employ roof distribution ducts, condensation will form on the ducts and outlet grilles when the doors are opened to discharge or take on passengers after the system is in operation. In time it may cause mildew and discoloration of the head lining material. This problem should be investigated.

COMPRESSOR DRIVE

Normally, people do not keep the engine running when they sit in a stationary car especially under warm weather conditions; however, with air conditioning they enjoy the comfort of keeping cool, and some trouble has been encountered as a result of lengthy operation of the cooling system with the engine idling. This practice results in high head pressure and high temperatures under the hood, both of which are unfavorable to the compressor belt drive. These conditions led to the adoption of steel core belts and later to a double-belt drive on two different makes of cars with which the author was associated. Due to design conditions, on a third model car the double-belt drive was used initially and it proved very satisfactory.

ENGINE COOLING

Air conditioning with the condenser located in front of the car radiator has added to the problem of cooling the engine, and again the longer periods of idling are one of the chief reasons. So-called heavy duty or export type radiators were used on some air conditioned cars, but while these radiators were better under heavy load medium and high speed running, they did not prove to be as good as standard radiators for the idling condition. Consequently, the standard low restriction domestic radiator is now extensively employed. A more efficient fan has been adopted which is driven at a slightly higher speed than the fans on cars which are not air conditioned.

ENGINE IDLING

To alleviate the problems which arise from longer periods of idling the engine two solutions have been tried. First, an effort has been made to educate the drivers of air conditioned cars to speed up the engine during idling; and second, cars with air conditioning units have been equipped with a device which automatically increases the engine idle speed when the gear shift lever is in neutral with the air conditioner on. The higher idling speed will not only improve the compressor drive and engine cooling condition; it will also result in better cooling from the air conditioner.

CONTROLS

The approach to the control problem has varied widely among the car manufacturers. Experience with heaters and similar devices has shown that the public will obtain the best results if the controls or adjustments are kept to an absolute minimum, especially those controls or adjustments which are accessible to the driver or the occupants of the car.

In the system with which the author has been associated there is only one control, namely the blower speed control, accessible to the occupants of the car.

The two outside air dampers and the minimum temperature setting of the automatic temperature control valve are only accessible by opening the door to the trunk.

On the other hand, one make of car has 18 controls or adjustments accessible to the driver and the occupants for controlling air velocity, air distribution, temperature, admission of outside air and for starting and stopping the system. Theoretically, the large number of controls permits adjusting the system so that each passenger in the car can have conditions to his particular liking. Actually, in many cases the control available to one individual may effect air or temperature distribution to the discomfort of others.

It is believed that a well-engineered system with simple controls in the hands of the individual operator will give better general comfort with fewer problems and better appearance. Simple controls also offer an excellent opportunity for cost reduction.

SERVICING THE SYSTEM

With all new devices on automobiles, there is a problem to get service parts in the hands of dealers and educate shop employees in servicing; automobile air conditioning systems have proved no exception. Many dealers have serviced the systems entirely in their own shops, while others have handled the mechanical work but called in refrigeration service companies to do the charging, testing and repairing of the refrigeration components.

The distribution of complete instruction books and service manuals can be very helpful to dealers for shop use, and factory trained personnel can be sent on occasion to assist in service problems. As a whole, troubles experienced with automobile air conditioning equipment have not been excessive and the dealers have been quite successful in handling maintenance and repairs.

FUTURE DEVELOPMENTS

Compressor: One of the most pressing needs of the automobile industry is a small, lightweight, durable compressor which will stand up under high speed operation and at the same time perform well at low speeds. Due to the styling trend, space under the hood of an automobile for mounting and driving a compressor is becoming more and more difficult to find. A smaller unit is also desirable so it will not interfere with the servicing of spark plugs or other engine parts. Such a compressor would require low torque to drive and a small pulley, possibly eliminating the need for two belts. It would also interfere less with engine balance and the compressor should be less subject to vibration. The compressor should be capable of dependable operation up to a speed of 6000 rpm, and it should be about ¼ to ⅓ the weight of present compressors.

Compressor Drive: Much study has been made on a variable-speed, quick-disconnect drive for the compressor. In addition, a variable speed drive would be very desirable for all of the engine driven accessories, namely: the water pump, fan, generator, pump for hydraulic steering, and the compressor. All of these units have an optimum speed that will meet the maximum performance requirement which is much lower than the maximum speed at which they are now driven. At the same time, it would be desirable to get better performance from these units at low engine speeds by driving them at higher speed. A variable speed accessory drive which would step up the speed of the accessory units at low engine speeds and then step down their speed at high engine speeds

would not only give better performance from the accessories, but it would also considerably reduce the maximum horsepower required to drive them.

Condenser Cooling: As the sale of automobile air conditioning increases, the styling and design of the car will have to be modified to make better provision for cooling the condenser without interfering with engine cooling. Possibly, this will be accomplished by having a separate fan or blower instead of depending on the present engine fan.

General Simplification: It is expected in the future that many of the present air conditioning systems will be greatly simplified and the novelty of having something to show the customer will decrease. It is obviously inconsistent to spend money for a wrap-around rear window and then cover up the wrap-around part with plastic ducts which are never really transparent and which detract from the appearance of the car.

In future systems there will be fewer controls for the passenger of the car to operate. A properly air conditioned car should have uniform temperature control with a minimum of drafts. It can be annoying to some of the passengers when each one is free to change the air distribution, and this is particularly true with children in the car.

Packaged Air Conditioning: Several companies have under development a so-called packaged air conditioning system in which all of the air conditioning equipment is assembled into a single unit which is mounted in the trunk of the automobile.

Advantages Claimed for Package Systems

1. The complete refrigeration system can be assembled and tested in a plant which specializes in air conditioning equipment, leaving only mechanical work to be performed in the automobile plant.

2. It lends itself to dealer installation and to transfer from an old car to a new one more readily than present systems.

3. Once the system is assembled and sealed it should be generally free of refrigeration type troubles.

4. It relieves the congestion in the engine compartment.

5. It will not interfere with engine cooling.

6. It is capable of giving maximum performance regardless of engine speed and consequently can be lower in capacity. This would be true only if the units are all electrically driven or driven by an auxiliary power plant.

7. The system can be completely inoperative when air conditioning is not required, and capacity and temperature can be controlled by varying the compressor speed, again depending on type of drive.

Disadvantages of Package System

1. It will take up more room in the trunk, and if a minimum of trunk space is to be used, it will have to be tailored to fit between the hump over the rear axle housing, the package shelf, the trunk lid hinges and the spare tire; so there will be little chance of being able to transfer the unit from one car to another, unless of the same model.

2. To cool the condenser and supply outside air for the evaporator will require a much larger outside air intake which involves a styling problem, as well as special openings in the body or trunk lid.

3. With the greater amount of machinery in the trunk, it will be much more difficult to seal off the noise from the passenger compartment.

4. If an electric drive is used, it calls for a much larger generator which may be either a-c or d-c and which must be driven from the engine, probably with special means for cooling. Along with the wiring and extra motors, this adds materially to cost. If an auxiliary gas engine is used, it presents a noise problem. If a hydraulic drive, the piping from pump to motors is bulky and the hydraulic system very costly. If a flexible cable drive from the engine is used, we run into most of the same problems that we have in the present compressor drive, plus additional problems of noise, whip and life of the flexible cables.

5. Experience with the package system up to the present time would indicate that it will cost more than the type of system now in use.

The past year has demonstrated convincingly that automobile air conditioning is now an important part of automobile manufacturing and sales, as indicated by public demand; and that the sale of many automobiles may depend on the dealers' ability to deliver the car with air conditioning.

Automobile manufacturers recognize its possibilities as a profitable accessory and as an aid to selling cars, so there is every reason to believe that in forthcoming cars, designers will make the necessary provisions for a good installation.

Whether the ultimate design will follow present practice or the packaged system type is difficult to determine now, but present indications point to lighter weight, less bulk, lower costs and simplification in the installation and operation of the system.

DISCUSSION

H. B. NOTTAGE, Encino, Calif. (WRITTEN): It is agreed that the matter of a compressor drive is an outstandingly important problem. In this connection it is proper to look more broadly into the future.

1. Small turbo-compressors can be designed with minimum requirements for space and weight. Indeed, such units are in established production for the air cycle and can be built for vapor cycles also.

2. Lack of a small driving turbine, fully developed and available at a competitive price is a major obstacle for turbo-compressors to be used in cars and trucks. Economic aspects are dominant and constitute the usual challenge to engineers.

3. Turbine-driven automobiles and trucks are on the way. Among these, one possibility is a turbine-driven alternator, with electric motors on the wheels. This combination could provide plenty of flexible auxiliary power. Another possible type would be a free-piston compressor for the power turbine, and this certainly could amply supply auxiliary turbine units.

Recognizing these points with their fullest implications, would the author wish to broaden his comments regarding the future for, say 5, 10 and 20 years?

AUTHOR'S CLOSURE: I assume that Dr. Nottage has only proposed the use of the small turbo-compressor type unit in connection with a so-called package type of air conditioning system in which all of the major units are located in the trunk of the automobile. The cost of these units would be high due to the accuracy required in making the various parts.

From our experience with this type of air conditioning system, even with the conventional drives now available, they are more expensive than the type of system where the compressor, the condenser and similar parts are located under the hood. The evaporator unit is a separate unit located either in the trunk or for the front end type of system under the hood but not necessarily combined in a package with the compressor, condenser, etc.

When the author is asked to comment regarding the future for say 5, 10 and 20 years, anything is possible. With the added knowledge which we will have in the years ahead, it may be that the turbo-compressor will be developed to the point where it is entirely practical for use in automobile air conditioning.



1496

No. 1496

PRESSURE LOSS OF AIR FLOWING THROUGH 45-DEGREE WOODEN LOUVERS

By P. R. COBB*, COLLEGE STATION, TEX.

THE RESISTANCE of louvers to air flow is important whenever any air moving device is to operate in conjunction with one or more sets of louvers. Most stationary louvers today are 45-deg louvers. Therefore, the pressure drop through this type of louver is of particular interest.

Despite the widespread use of louvers in applications such as shielding devices for light, wind and rain; cooling tower walls; and to cover ventilation ports in homes and buildings, there seems to be little published material on louver resistance to air flow. The first results from a research program of the Texas A. and M. College Engineering Experiment Station evaluating the effect of different variables on pressure loss in 45-deg wooden louvers are therefore presented†.

EQUIPMENT

The test equipment illustrated in Figs. 1 and 2 consisted of a fan, plenum chamber, and inlet duct. The 24-in. centrifugal fan driven by a 2 hp motor discharged air through the duct into the plenum chamber. This chamber, 7 ft high, 11 ft long, and 7 ft wide, contained five concentric cone diffusers designed to distribute the air uniformly and to give to the chamber the characteristics of a large room. Connected to the exhaust side of the plenum chamber was a duct having a rounded entrance and measuring 3 sq ft in area and 4.5 ft in length. The various louvers tested were attached to the outlet end of this duct.

The air flow rate was controlled by vanes on the inlet side of the fan. Air flow measurements were made in one of two 11 ft long, 13 in. diameter ducts used for connecting the fan to the plenum chamber. One of these ducts con-

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tained a 6 in. throat diameter nozzle which was used for measuring low rates of air flow. The nozzle manometer was connected between an upstream static tap in the duct and a tap in the throat of the nozzle. The second duct was equipped for Pitot tube measurements and was used for the higher rates of flow. In the 3 sq ft inlet duct to the louver, a static pressure tap was located 2 ft in front of the louver. This tap was connected to the pressure side of a zero-to-one inch inclined scale manometer. The other side of the manometer was vented to the atmosphere. The pressure indicated on this manometer was the static pressure loss of the air flowing through the louver.

Barometric pressure was read from a mercury column barometer and corrected to the standard conditions of the U. S. Department of Commerce Weather Bureau, namely 45 deg latitude, sea level, and 32 F temperature.

Temperature measurements were made with two 0 to 120 F mercury-in-glass thermometers. The thermometers were placed at the intake side of the fan, one indicating the dry-bulb temperature and the other the wet-bulb temperature.

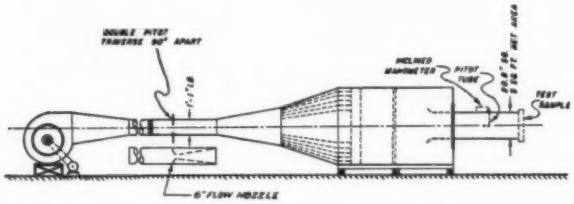


FIG. 1. ARRANGEMENT OF TEST EQUIPMENT FOR RESISTANCE MEASUREMENTS

The louvers (Fig. 2) were made up of boards that had a planed surface and were 20.78 in. long and 0.81 in. thick. The edge of each louver board was beveled to a 45-deg angle, thus making the leading surface of each board perpendicular to the air stream. By changing the number of boards in each louver from three to 17, the free area ratio was varied from 0.8893 to 0.1148. All louvers had the same overall width, 4.37 in. in the direction parallel to the flow of incoming air.

EXPERIMENTATION

Because of the relative ease of data calculation, the flow nozzle method of air flow measurement was used whenever possible. As the free area increased, fewer test runs were possible with this method. The louver with the largest free area permitted only three test runs to be made by the nozzle method while the smallest free area test was made entirely by the nozzle method. One test was run completely with the nozzle and then over again with the Pitot traverse. The difference in the results was found to be negligible (see Table 1).

When the nozzle was used for measurement of air flow, two sets of temperature and pressure drop readings were made after the system reached steady flow. When the Pitot tube was used for measuring air flow, three sets of temperature and pressure drop readings were obtained—one at the beginning of the test when

the system had reached steady flow, one halfway through the Pitot tube traverse, and one at the end of the traverse. Averages obtained from these readings were used in subsequent calculations.

Each louver was tested at different rates of air flow adjusted to give increments of pressure drop of about 0.1 in. of water. The number of tests on a particular louver depended upon the free area and the maximum rate of air flow available. The number varied from five for louvers with large free area to nine for louvers having small free area.

All manometers used in the test setup were calibrated with a micro-manometer and a hook gage to 0.001 in. of water before being used. In the calculation of data these corrections were applied to all final manometer readings.

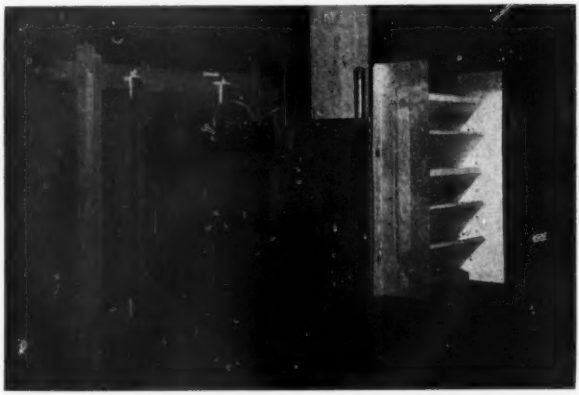


FIG. 2. INLET DUCT AND LOUVERS FOR TESTING

The velocity of approach air to the louvers was tested for uniformity of velocity front. The results of this test showed a velocity front with less than 5 percent variation throughout the center one-half of the test section. The velocity maintained this front until very close to the wall of the inlet duct where it dropped off sharply.

Leakage throughout the system was checked and found to be less than 1 percent of the air delivered by the blower. The method used to measure leakage was that outlined in a Texas Engineering Experiment Station Research Report[1].

ANALYSIS AND COMPUTATION OF DATA

Dimensional Analysis: The loss of pressure of air flowing through a louver obviously depends upon the dimensions of the louver, the quantity of air flowing, and upon certain properties of the air such as density and viscosity. At constant atmospheric pressure, density and viscosity are functions chiefly of the temperature of the air. For this analysis, seven possible variables were considered. These are listed with their respective symbols and dimensions.

[1] Exponent numerals refer to References.

A'/A = ratio of free area to face area, dimensionless.
V = velocity of the approach air to the louver, Length/Time.
μ = viscosity of air flowing, Mass/(Length) (Time).
ρ = density of air flowing, Mass/(Length)^3.
P = pressure loss of air flowing tnrough the louver, Mass/(Time)^2 (Length).
R = hydraulic radius of the louver, Length.
ϕ = angle made with the vertical by the center line of a louver board, dimensionless.

Using dimensional analysis[2] to group the variables, it was found that there are four dimensionless groups satisfying the Buckingham Pi Theorem. These four groups are:

$$\pi_1 = P/\rho V^2$$
$$\pi_2 = RV\rho/\mu$$
$$\pi_3 = A'/A$$
$$\pi_4 = \phi$$

In this investigation the louver board slope, or the angle Φ, was constant (45-deg), so π_4 was the same for all tests. The free area ratio was varied by changing the number of louver boards so that π_3 varied from 0.1148 to 0.8893.

Calculation of Velocity: The approach of the air entering the louver was calculated by first finding the volume rate of flow through the system and then dividing this rate by the area of the inlet duct. This gave the mean velocity of approach.

The volume rate of flow of air through the system was calculated for both the nozzle and Pitot tube method by means of the equation[3]

$$Q = 1096.5\,KA\sqrt{\frac{h_v}{\rho}}$$

where

K = the coefficient of discharge multiplied by the velocity of approach correction factor (*for the nozzle*).
K = a traverse correction factor (*for the Pitot tube*).
h_v = velocity pressure, inches of water.
A = face area or total area covered by the louver in a plane perpendicular to the approach air, square feet.
ρ = the density of the air flowing, pounds mass per cubic foot.

The velocity pressure at the throat of the nozzle was read directly from an inclined scale manometer. For the Pitot tube, the velocity pressure used in the equation was the square of the average of the square roots of the traverse readings. The traverse consisted of 20 readings, 10 taken along a horizontal axis and 10 along a vertical axis of the duct and at the center of equal areas, according to recommended practice[3].

Density: The density of the air was obtained from a circular air density slide rule[4] developed by the Texas Engineering Experiment Station using the observed barometric pressure and the dry- and wet-bulb temperatures of the air.

The density of the room air for all tests ranged from 0.06752 to 0.07278, and depended upon normal ambient room conditions, except that the lowest density (0.06752) was obtained by warming the room artificially to about 110 F by means of a gas-fired furnace.

Hydraulic Radius: The hydraulic radius (R) was calculated from the dimensions of the louver. By changing the number of louver boards the value of the hydraulic radius was varied from 0.0097 ft to 0.266 ft.

Viscosity: The viscosity of the air was obtained from published tables[5]. Values of viscosity ranged from 123×10^{-7} to 127×10^{-7} pounds mass per ft sec.

Conversion Factors: In calculating the dimensionless groups $P/\rho V^2$ and $RV\rho/\mu$ from the observed and calculated quantities P_L, ρ, V, R, and μ it was necessary, because of inconsistent units, to use conversion factors.

P_L was expressed in inches of water, ρ in pounds mass per cubic foot, and V in feet per minute. The group $P_L/\rho V^2$ then had the dimensions inches of water minutes squared feet per pounds mass. The group, with these dimensions, was then multiplied by 6×10^5 in order to make the group dimensionless.

Similarly, R was expressed in feet, V in feet per minute, ρ in pounds mass per cubic foot and μ in pounds mass per foot per second. The group $RV\rho/\mu$ then had the dimensions seconds per minute and was divided by 60 in order to make the group dimensionless.

Graphical Representation of Data: It might be expected that for a given louver the most important variables affecting pressure drop are density and velocity, each of which appear in the dimensionless numbers of π_1 and π_2. In analyzing the results of the tests, values of π_1 and π_2 were calculated from observed data and then plotted against each other (Fig. 3) to give a family of curves having π_3 as a parameter, π_4 being constant for all tests. From this plot equations were obtained giving the relationship between pressure drop, velocity, density, and free area ratio.

ANALYSIS OF GRAPHICAL REPRESENTATION

From the plot of $P/\rho V^2$ against $RV\rho/\mu$ (Fig. 3) for values of the free area ratio above 0.5, the value of $P/\rho V^2$ for each louver tested was approximately constant (see Table 1). Below this value of free area ratio the different lines representing each louver were still straight, but as the free area ratio decreased the slope of the lines decreased.

NOMENCLATURE

A = face area or total area covered by the louver in a plane perpendicular to the approach air, square feet.
A' = free area or area in the louver open to the flow of air in a plane perpendicular to the approach air, square feet.
A'/A = ratio of free area to face area, dimensionless.
g = local acceleration due to gravity, approximately 32.11 feet per (second) (second).
P = pressure loss of air flowing through the louver, pounds mass per foot minute squared = ($6 \times 10^5 \times$ pressure in inches of water*).
P_L = pressure loss of air flowing through the louver, inches of water.
P_{L_s} = pressure loss of air flowing through the louver, corrected to a standard air density of 0.075 pounds mass per cubic foot, inches of water.
R = hydraulic radius of the louver, feet. The free area divided by the perimeter of the opening in a plane perpendicular to the air flow.
V = average velocity of the air flowing through the inlet duct, feet per minute.
ρ = density of air flowing, pounds mass per cubic foot.
ρs = density of standard air, 0.075 pounds mass per cubic foot.
μ = viscosity of air flowing, pounds mass per foot second.
ϕ = angle made with the vertical by the center line of a louver board, dimensionless.

* A unit of pressure equivalent to the pressure exerted by a 1 in. column of water to a density of 62.3 pounds mass per cubic foot.

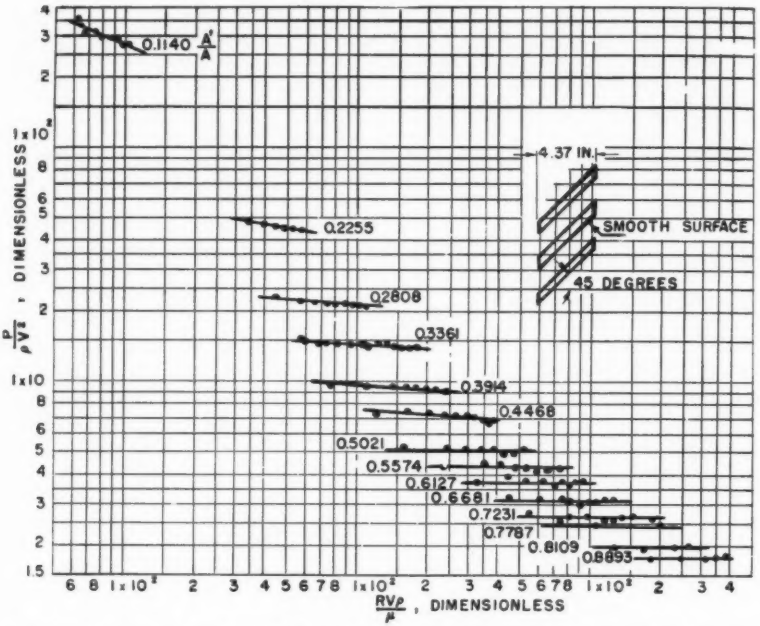


FIG. 3. VARIATION OF $P/\rho V^2$ WITH $RV\rho/\mu$ FOR DIFFERENT VALUES OF RATIOS FOR FREE AREA TO FACE AREA

For any test in the area of Fig. 3 comprising tests with a free area ratio greater than 0.5, a change in the group $RV\rho/\mu$ would not change the value of $P/\rho V^2$ for that test. This means that in this region $P/\rho V^2$ was independent of a change in $RV\rho/\mu$. Hence, the pressure loss was independent of the combined effect of viscosity of the air and hydraulic radius of the louver section since this was the only group in which μ and R appeared.

With $P/\rho V^2$ independent of $RV\rho/\mu$ for 45-deg louvers with a free area ratio greater than 0.5 it was possible to graph $P/\rho V^2$ against A'/A with only the angle as a parameter. This led to the functional relationship,

$$P/\rho V^2 = f\,(A'/A) \quad \ldots\ldots\ldots\ldots\ldots \quad (1)$$

When these constant values of $P/\rho V^2$ and the corresponding values of A'/A were plotted on log paper a straight line was obtained, the equation of which was $P/\rho V^2 = a(A'/A)^b$. Using the slope intercept formula solving for a and b, the equation of the straight line expressing the functional relationship was:

$$P/\rho V^2 = 1.486\,(A'/A)^{-1.783} \quad \ldots\ldots\ldots\ldots \quad (2)$$

This is a general equation, applicable in the region of the variables tested, relating the pressure drop, velocity, free area ratio, and density of air flowing through louvers with a free area ratio above 0.5.

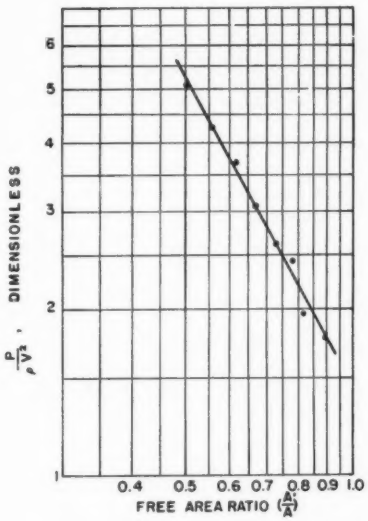


FIG. 4. VARIATION OF $P/\rho V^2$ WITH RATIOS OF FREE AREA TO FACE AREA

By multiplying both sides of Equation 2 by the density of standard air and a conversion factor, the special equation for the pressure loss in inches of water can be shown to be:

$$P_L' = 1.855 \times 10^{-7}\, V^2\, (A'/A)^{-1.783} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

For values of free area ratio below 0.5, the pressure loss at given conditions can be determined by calculating the value of the group $RV\rho/\mu$, and by finding the corresponding value of $P/\rho V^2$ for the particular free area ratio. The pressure loss may then be easily computed by using the value from the graph and the value of the density and velocity.

An equation for the sloping lines in the region of free area ratios below 0.5 was not written because of its cumbersome nature. The equation in functional form would appear as:

$$P/\rho V^2 = f_1\,(A'/A)\, f_2\,(RV\rho/\mu)\, f_3\,(A'/A) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Not only would this equation present difficulties in solving, but also, most louvers designed today have a free area ratio above 0.5. It is recommended that for free area ratios below 0.5 the graph be used, and for free area ratios above 0.5 the equation be used.

RESULTS AND CONCLUSIONS

The results of this research were the determination of the principal variables and their influence on the pressure drop of air flowing through 45-deg wooden louvers.

1. The principal variable affecting pressure loss in 45-deg, wooden louvers with a free area ratio above 0.5 is the average velocity of the approach air. It can be seen

TABLE 1—TABULATION OF DATA AND CALCULATIONS ON TESTS OF PRESSURE LOSS OF AIR FLOWING THROUGH 45-DEG WOODEN LOUVERS

$\frac{\pi_3}{A'/A}$	P_L	ρ	h_v		V	V^2	$\frac{\pi_1}{P/\rho V^2}$	R	μ	$\frac{\pi_2}{RV\rho/\mu}$
Dimension-less	In. of water	lb_m/ft^3	In. of water		ft/min	ft^3/min^2	Dimension-less	ft	lb_m/ft sec	Dimension-less
0.8893	0.110	0.07154	[a]N	7.106	722.4	521,900	1.764	0.2665	124×10^{-7}	1.851×10^4
	0.198	0.07145		12.719	967.1	953,300	1.774			2.475×10^4
	0.308	0.07145	[b]P	0.956	1220.	1,487,000	1.735			3.123×10^4
	0.397	0.07160		1.194	1361.	1,853,000	1.792			3.490×10^4
	0.501	0.07138		1.486	1522.	2,315,000	1.815			3.892×10^4
0.8109	0.127	0.07206	[a]N	7.272	728.1	530,100	1.987	0.1832	123×10^{-7}	1.292×10^4
	0.211	0.07206		12.587	958.1	917,900	1.914			1.700×10^4
	0.407	0.07205	[b]P	1.112	1310.	1,715,000	1.971			2.324×10^4
	0.497	0.07187		1.342	1441.	2,076,000	1.995			2.550×10^4
	0.548	0.07195		1.466	1505.	2,265,000	2.013			2.666×10^4
0.7787	0.098	0.06964	[a]N	4.423	577.7	333,700	2.528	0.1411	128×10^{-7}	7.448×10^3
	0.199	0.06980		9.296	836.6	699,800	2.430			10.81×10^3
	0.295	0.06975		13.965	1026.	1,052,000	2.405			13.25×10^3
	0.392	0.06961	[b]P	0.876	1183.	1,400,000	2.402			15.25×10^3
	0.485	0.06950		1.062	1303.	1,698,000	2.457			16.77×10^3
	0.591	0.06945		1.252	1416.	2,005,000	2.534			18.21×10^3
	0.694	0.06952		1.492	1545.	2,386,000	2.499			19.88×10^3
0.7234	0.098	0.07005	[a]N	4.023	549.3	301,700	2.760	0.1100	127×10^{-7}	5.550×10^3
	0.202	0.07009		8.750	809.7	655,700	2.632			8.188×10^3
	0.292	0.07008		12.595	971.7	944,100	2.636			9.823×10^3
	0.395	0.07024	[b]P	0.841	1154.	1,331,000	2.523			11.69×10^3
	0.493	0.07037		1.020	1269.	1,611,000	2.600			12.89×10^3
	0.584	0.07035		1.184	1367.	1,871,000	2.652			13.87×10^3
	0.706	0.07015		1.414	1497.	2,241,000	2.686			15.15×10^3

TABLE 1—TABULATION OF DATA AND CALCULATIONS ON TESTS OF PRESSURE LOSS OF AIR FLOWING THROUGH 45-DEG WOODEN LOUVERS *(continued)*

TABLE 1—TABULATION OF DATA AND CALCULATIONS ON TESTS OF PRESSURE LOSS OF AIR FLOWING THROUGH 45-DEG WOODEN LOUVERS *(continued)*

0.6681	0.114	0.07200	[a]N	4.047	543.4	295,300	3.211	0.0868	123×10^{-7}	4.563×10^3
	0.202	0.07200		7.305	730.3	533,300	3.156			6.140×10^3
	0.309	0.07215		11.362	909.7	827,500	3.106			7.657×10^3
	0.347	0.07186		12.942	972.7	946,200	3.058			8.152×10^3
	0.415	0.07186	[b]P	0.748	1076.	1,158,000	2.984			9.020×10^3
	0.502	0.07200		0.893	1175.	1,380,000	3.026			9.868×10^3
	0.600	0.07204		1.049	1272.	1,618,000	3.084			10.69×10^3
	0.718	0.07210		1.241	1384.	1,914,000	3.116			11.64×10^3
	0.846	0.07205		1.434	1488.	2,213,000	3.179			12.51×10^3
0.6127	0.111	0.07115	[a]N	3.358	498.0	248,000	3.764	0.0697	124×10^{-7}	3.292×10^3
	0.206	0.07115		6.134	673.0	452,900	3.828			4.449×10^3
	0.295	0.07110		8.960	813.6	662,000	3.755			5.374×10^3
	0.405	0.07110		12.475	960.3	922,100	3.699			6.343×10^3
	0.493	0.07119	[b]P	0.751	1083.	1,172,000	3.540			7.162×10^3
	0.582	0.07110		0.871	1167.	1,363,000	3.606			7.706×10^3
	0.684	0.07105		1.012	1259.	1,584,000	3.642			8.309×10^3
	0.784	0.07105		1.133	1331.	1,772,000	3.731			8.785×10^3
	0.909	0.07108		1.296	1424.	2,027,000	3.779			9.337×10^3
0.5574	0.099	0.07150	[a]N	2.588	436.1	190,200	4.357	0.0564	124×10^{-7}	2.366×10^3
	0.227	0.07150		5.747	649.8	422,300	4.495			3.522×10^3
	0.310	0.07146		8.082	761.6	580,000	4.477			4.127×10^3
	0.416	0.07146		11.075	902.4	814,300	4.280			4.889×10^3
	0.467	0.07170		12.588	960.4	922,400	4.225			5.218×10^3
	0.581	0.07145	[b]P	0.761	1088.	1,184,000	4.112			5.892×10^3
	0.709	0.07150		0.917	1194.	1,425,000	4.166			6.471×10^3
	0.788	0.07168		1.019	1257.	1,580,000	4.165			6.829×10^3
	0.931	0.07179		1.189	1356.	1,841,000	4.217			7.378×10^3
0.5021	0.089	0.06997	[a]N	1.923	380.0	144,400	5.247	0.0458	127×10^{-7}	1.596×10^3
	0.201	0.06995		4.370	572.9	328,200	5.231			2.406×10^3
	0.298	0.06994		6.564	702.2	493,100	5.158			2.950×10^3
	0.395	0.06989		8.817	813.9	662,500	5.103			3.417×10^3
	0.493	0.06990		11.075	912.2	832,200	5.064			3.831×10^3
	0.587	0.06988	[b]P	0.655	1021.	1,042,000	4.815			4.285×10^3
	0.700	0.06984		0.765	1104.	1,217,000	4.923			4.631×10^3
	0.791	0.06985		0.859	1169.	1,367,000	4.953			4.906×10^3
	0.891	0.06980		0.952	1231.	1,516,000	5.033			5.160×10^3
	0.971	0.06980		1.036	1285.	1,650,000	5.040			5.386×10^3

[a] N Using 6 in. flow nozzle. [b] P Using 2, ten point Pitot traverses. $\pi_4(\phi)$ 45 deg for all tests.

TABLE 1—TABULATION OF DATA AND CALCULATIONS ON TESTS OF PRESSURE LOSS OF AIR FLOWING THROUGH 45-DEG WOODEN LOUVERS *(continued)*

$\frac{\text{Eq 2}}{A'/A}$	P_L	ρ	h_v		V	V^2	$\frac{\text{Eq 1}}{P/\rho V^2}$	R	μ	$\frac{\text{Eq 3}}{RV\rho/\mu}$
Dimensionless	In. of water	lb m/ft³	In. of water		ft/min	ft²/min²	Dimensionless	ft	lb m/ft sec	Dimensionless
0.4468	0.108	0.07060	[a]N	1.685	354.2	125,400	7.276	0.0370	126x10⁻⁷	1.224x10⁵
	0.198	0.07059		3.039	475.6	226,200	7.399			1.644x10⁵
	0.297	0.07055		4.669	589.7	347,700	7.248			2.037x10⁵
	0.407	0.07055		6.451	693.1	480,400	7.182			2.394x10⁵
	0.504	0.07055		8.013	772.6	596,900	7.157			2.669x10⁵
	0.613	0.07054		9.905	858.9	737,700	7.040			2.966x10⁵
	0.691	0.07055		11.248	915.2	837,500	6.991			3.161x10⁵
	0.800	0.07025	[b]P	0.648	1013.	1,025,000	6.648			3.483x10⁵
	0.889	0.07020		0.717	1066.	1,136,000	6.668			3.663x10⁵
	0.968	0.07019		0.769	1104.	1,218,000	6.772			3.793x10⁵
0.3914	0.094	0.06984	[a]N	1.108	288.7	83,330	9.615	0.0297	127x10⁻⁷	7.851x10⁴
	0.186	0.06984		2.220	408.7	167,000	9.551			11.12 x10⁴
	0.296	0.06984		3.525	515.0	265,200	9.543			14.01 x10⁴
	0.395	0.06984		4.765	598.8	358,500	9.434			16.29 x10⁴
	0.484	0.06983		5.818	661.7	437,800	9.461			18.00 x10⁴
	0.591	0.06983		7.198	736.0	541,700	9.346			20.03 x10⁴
	0.689	0.06983		8.491	799.3	638,900	9.226			21.75 x10⁴
	0.788	0.06987		9.880	861.9	742,900	9.077			23.46 x10⁴
	0.877	0.06980		10.944	907.6	823,800	9.177			24.69 x10⁴
0.3361	0.137	0.06802	[a]N	1.035	282.8	79,950	15.09	0.0236	130x10⁻⁷	5.820x10⁴
	0.191	0.06775		1.473	337.9	114,200	14.73			6.926x10⁴
	0.278	0.06740		2.178	411.8	169,800	14.50			8.397x10⁴
	0.404	0.06762		3.186	497.5	247,500	14.44			10.18 x10⁴
	0.475	0.06765		3.772	541.2	293,000	14.34			11.08 x10⁴
	0.602	0.06765		4.767	608.5	370,200	14.37			12.45 x10⁴
	0.696	0.06680		5.561	661.4	437,400	14.26			13.37 x10⁴
	0.783	0.06740		6.318	701.8	492,500	14.11			14.31 x10⁴
	0.880	0.06750		7.142	745.6	555,900	14.03			15.23 x10⁴
	0.988	0.06739		8.070	793.2	629,200	13.93			16.17 x10⁴

TABLE 1—TABULATION OF DATA AND CALCULATIONS ON TESTS OF PRESSURE LOSS OF AIR FLOWING THROUGH 45-DEG WOODEN LOUVERS *(concluded)*

TABLE 1—TABULATION OF DATA AND CALCULATIONS ON TESTS OF PRESSURE LOSS OF AIR FLOWING THROUGH 45-DEG WOODEN LOUVERS *(concluded)*

0.3361	0.133	0.07085	[a]N	1.014	274.2	75,180	14.89	0.0236	124×10^{-7}	6.114×10^{2}
	0.190	0.07080		1.483	331.7	110,000	14.59			7.389×10^{2}
	0.308	0.07080		2.426	424.3	180,000	14.44			9.449×10^{2}
	0.385	0.07075		3.032	474.5	225,200	14.45			10.56×10^{2}
	0.514	0.07074		4.056	548.9	301,300	14.44			12.21×10^{2}
	0.594	0.07081		4.704	590.8	349,000	14.37			13.16×10^{2}
	0.672	0.07079		5.352	630.3	397,200	14.29			14.04×10^{2}
	0.789	0.07071		6.358	687.3	472,400	14.13			15.29×10^{2}
	0.891	0.07067		7.267	734.9	540,100	13.98			16.34×10^{2}
	0.979	0.07065		8.054	774.0	599,000	13.85			17.20×10^{2}
0.2808	0.197	0.06994	[a]N	0.992	272.9	74,500	22.63	0.0183	127×10^{-7}	4.583×10^{2}
	0.295	0.06994		1.528	338.8	114,800	21.95			5.686×10^{2}
	0.395	0.06994		2.091	396.3	157,100	21.51			6.649×10^{2}
	0.492	0.06994		2.609	444.2	197,300	21.31			7.460×10^{2}
	0.593	0.06994		3.151	486.5	236,700	21.39			8.169×10^{2}
	0.685	0.07000		3.665	524.5	275,100	21.27			8.811×10^{2}
	0.791	0.07000		4.232	563.6	317,700	21.25			9.470×10^{2}
	0.887	0.07000		4.767	598.1	357,800	21.17			10.05×10^{2}
	0.986	0.07000		5.364	634.5	402,600	20.92			10.66×10^{2}
0.2255	0.380	0.07150	[a]N	0.903	257.6	66,360	47.95	0.0137	124×10^{-7}	3.400×10^{2}
	0.507	0.07144		1.235	301.4	90,840	46.76			3.962×10^{2}
	0.615	0.07144		1.526	335.0	112,200	45.94			4.408×10^{2}
	0.731	0.07144		1.837	367.5	135,100	45.36			4.832×10^{2}
	0.832	0.07144		2.117	394.6	155,700	44.80			5.189×10^{2}
	0.873	0.07144		2.243	406.2	165,000	44.40			5.341×10^{2}
	0.992	0.07144		2.557	433.6	188,000	44.23			5.704×10^{2}
0.1148	0.308	0.07271	[a]N	0.100	85.13	7,248	349.9	0.00617	123×10^{-7}	6.367×10
	0.462	0.07271		0.160	107.4	11,550	329.4			6.531×10
	0.577	0.07257		0.213	124.5	15,450	308.1			7.555×10
	0.705	0.07257		0.275	141.1	19,910	292.2			8.564×10
	0.870	0.07278		0.357	160.4	25,740	278.2			9.764×10
	0.954	0.07278		0.403	170.7	29,120	269.5			10.39×10
0.1148	0.473	0.07168	[a]N	0.169	111.2	12,350	319.9	0.00617	124×10^{-7}	6.665×10
	0.626	0.07168		0.239	132.3	17,500	299.1			7.930×10
	0.813	0.07171		0.329	155.2	24,110	281.6			9.308×10
	0.970	0.07171		0.406	172.4	29,730	272.6			10.34×10

[a] N Using 6 in. flow nozzle. [b] P Using 2, ten point Pitot traverses. $\pi_4(\phi)$ 45 deg for all tests.

from Equation 2, that with this type of louver the pressure loss varies as the square of the velocity.

2. The constant of proportionality between the pressure loss in inches of water referred to standard air and the velocity squared was equal to

$$1.855 \times 10^{-7} (A'/A)^{-1.783}$$

This dimensional constant was based on the slope and intercept of the straight line in Fig. 4.

3. The combined effect of the viscosity of the air and the hydraulic radius of the louver for 45-deg, wooden louvers with a free area ratio above 0.5 was found to be negligible. This conclusion was reached after the dimensionless group $P/\rho l'^2$ was plotted as a constant for different values of the free area ratio.

The decreasing slope of the lines in Fig. 3 indicates that as the free area ratio decreased, the combined effect of the viscosity and the hydraulic radius on the pressure drop increased.

4. The decreasing slope of the lines in Fig. 3 also seems to indicate that the pressure loss approaches a linear function of the velocity as the free area ratio decreases.

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4. Determination of the Density of Moist Air in Fan Performance Calculations, by Guy Johnson, Jr. (Texas Engineering Experiment Station, College Station, Tex., 1953).

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DISCUSSION

S. F. GILMAN, Syracuse, N. Y. (WRITTEN): In the analysis section it is stated that the pressure loss *obviously depends upon the dimensions of the louver,* which means the geometry of the system. However, in the selection of the principal variables affecting the pressure loss and in the resultant dimensionless groups obtained by dimensional analysis, the geometry of the system remains unspecified. The free area fixes only the number of louvers and the angle ϕ, and these do not fix the geometry.

The dimensionless groups actually apply only to the condition that geometrical similarity is maintained. That this condition is not fulfilled is easily seen by comparing the 3 louver and the 17 louver configurations. With only 3 louvers, the vertical distance between louvers, or what could be called the slot height, is more than 9 in. so that the length of the channel through which the air passes is less than one slot height.

On the other hand, with 17 louvers there are a series of channels having a slot height less than 3/16 in. and a length of more than 40 slot heights. These are far from being geometrically similar; *i.e.,* far from having the same shape. Moreover,

for such a wide range of slot heights the relative roughness of the channels changes considerably. Therefore, the dimensional analysis represents an oversimplification of the flow conditions. A characteristic dimension of the louver and the absolute roughness should have been included.

An additional figure in the paper showing the actual configuration of the louvers for at least 4 area ratios would have been valuable in analyzing some practical aspects. For example, since the 17 louver condition results in each louver board overlapping the inlets of the 4 boards above it, I doubt that the smaller area ratio configurations have any real practical significance.

In Fig. 3, it is seen that the pressure loss is independent of Reynolds number down to an area ratio of about 0.5. This means simply that the inertia forces are predominant, a fact which would be anticipated from the geometry. Below 0.5, the channel becomes long in comparison with the slot height and the influence of viscous forces becomes increasingly significant. This is shown by the variation in loss coefficient with Reynolds number for a constant area ratio. Here also we see a remarkable thing. Taking the area ratio of 0.2808 as an example, the curve indicates only a relatively small influence of viscous forces over a Reynolds number range from 460 to 1100. This is contrary to my expectation that viscous forces should be predominant below a Reynolds number of about 2300; in fact, for such low Reynolds numbers I would expect the flow to be laminar. If it were, the slopes of the constant area lines would be proportional to $64/R$ and consequently much steeper.

To resolve this contradiction to expectation, let us look at the author's definition of Reynolds number. Here we see that the hydraulic radius is based on the free area at the entrance to the louver, but the velocity refers to that in the inlet duct upstream. Recalling that the Reynolds number is the ratio of inertia to viscous forces, it is evident the definition of Reynolds number used in the paper is some viscous-inertia force combination obtained from two distinct regions of the flow. My questions are, therefore:

1. Is this really a Reynolds number at all in the true sense of the definition?

2. What particular significance does the author attach to such a definition of Reynolds number?

It seems to me that more importance is given to the variable V, the average velocity in the inlet duct, than it deserves. Moreover, as will be demonstrated, it overcomplicates the final results. Now certainly the pressure loss of a louver will be influenced by the flow conditions within the louver; therefore, the velocity V' should logically appear in the Reynolds number and loss coefficient axes of Fig. 3. We convert to such a basis by multiplying the $P/\rho V^2$ values by the square of the area ratio and dividing the Reynolds number values by the area ratio. When this is done, a not-so-surprising thing happens. Within experimental accuracy the 10 curves in Fig. 3 for area ratios between 0.8893 and 0.3914 are reduced to the single value of 1.4. This means that the deviations from geometrical similarity for these 10 configurations have an insignificant effect on the flow conditions, and therefore the pressure loss for all 10 area ratios can be expressed as 1.4 velocity heads referred to the mean velocity at the louver inlet slots. This not only simplifies the present Fig. 3 considerably but in addition eliminates the need for Fig. 4.

I do not agree with the author's conclusion that the velocity of the approach air is the principal variable affecting the pressure loss. The principal variables are the flow conditions within the louver boards, and not the conditions of the approaching flow. The fact that the two are connected through the continuity equation is not justification for the conclusion that V, instead of V' is of significance.

In summary, for all really practicable louver configurations the results of this paper can be expressed as 1.4 velocity heads referred to the mean velocity at the inlet slots, and this is all the designer needs to know.

F. W. RABE, Dallas, Tex. (WRITTEN): The author refers to louvers on cooling tower walls. It is assumed, therefore, that pressure loss determinations as given by Equation 3, for instance, apply to cooling tower louvers. A common application of louvers for cooling towers are the air intake-louvers of a conventionally designed counterflow induced draft tower. These louvers are usually located between the tower packing and the water basin. The air which leaves these louvers is turned and deflected approximately 135 deg before it enters the tower packing. If such louvers were provided with an inlet duct, similar to the one used by the author in the test equipment, the direction of the air would be changed both at the entrance and at exit from the louver. Air flowing through the louvers tested was deflected only 45 deg.

It is assumed that nearly uniform air velocities were maintained at all points of the louvers for a given test condition. In practice it is found that the air velocity through cooling tower louvers, such as described, is not uniform. The velocity of the air near the top of the louver is, in some cases, considerably higher than the velocity near the bottom. The reason for this difference in air velocities becomes obvious when the flow pattern of the air in a cooling tower, as outlined, is carefully analyzed. Average readings of louver air velocities, taken at a large cooling tower, varied from 963 fpm at the top to 353 fpm at the bottom of the louver. Readings taken midway between top and bottom indicated air velocities intermediate to those cited.

Conditions under which the louver tests were made did not assimilate air-flow conditions through cooling tower louvers as discussed. It is, therefore, doubtful if the determination of pressure loss as, for instance, given by Equation 3 applies to louvers if installed on a cooling tower as described. The air velocity V as used in Equation 3 denotes the average velocity of air through the inlet duct. Few, if any, louvers are equipped with such inlet ducts and it would, therefore, be of interest to know what effect the addition of an inlet duct has on the air-flow resistance.

There is a definite need for practical and reliable data giving air pressure losses through louvers. Considering this paper from a practical engineering point of view, it would seem that too much space is devoted to test procedure and test data and too little to the gist of the problem. These comments are in no way intended to detract from the value of the paper, which it is hoped will stimulate others to continue this investigation. The author is to be commended for the work he has done.

H. B. NOTTAGE, Encino, Calif. (WRITTEN): In order to broaden and expedite use of the data presented, it is suggested that the author should further clarify how the groups π_2, π_3, and π_4 include *all* the details of louver geometry and yet are the *best* independent moduli for correlation.

Such a clarification should save reader time in applying the data to other designs. In this regard, how far does the author feel that one might extend the data presented in Fig. 3 beyond the particular test range? If such an extension cannot be made at all, then why use dimensionless moduli?

R. W. MCKINLEY, Pittsburgh, Pa.: Although the author has indicated that this study was inspired by a concern for fan performance, references were made in the published preprint to the correlation with natural ventilation and to the type of louvers that are used frequently for daylight control, shading and solar heat reduction on the exterior of buildings.

Because this is a very interesting and important practical problem in the school design field today, I want to ask the author some questions regarding the application of this particular data to that type of problem.

For example: do these data apply equally well when the cross-section of the louver is very small as compared with the other dimensions, for example to the familiar metal slat venetian blind?

A second question is: what effect, if any, would the curvature of thin metal slat venetian blinds have on these data?

Perhaps you are familiar with the very interesting report, also from the Texas Engineering Experiment Station (No. 36 of July, 1952) on the subject of Geometry of Classrooms as related to Natural Lighting and Natural Ventilation. In this report they plotted some rather interesting natural ventilation flow diagrams with air velocity figures included. I would like the author to explain if and how these data on louver performance can be applied directly to such air velocity figures in a practical way so that one might determine what effect the application of louvers on the outside of natural ventilation inlet and exhaust areas would have on air flow velocity.

C. W. BEVIER*, College Station, Tex.: I agree with Dr. Gilman that the variables considered by the author cannot completely specify the geometry. In the work that I have done on louvers I have defined the geometry of the louver by the use of five shape dimensions (see Fig. A). These are the width of the louver board (the important one not considered by Mr. Cobb), the length of the louver board, the thickness of the louver board, the spacing of the lover boards measured perpendicular to the louver board, and the distance between louver boards measured at the entrance to the louver. Also the shape of the louver board must be maintained.

The louver shape that Mr. Cobb tested may be called a standard louver. This louver may be inverted so that the sharp edge is parallel to fluid flow and as a result forms what I call an inverted louver. Another shape may be obtained by splitting Mr. Cobb's louvers diagonally (in a cross-sectional view). This type of louver I refer to as a diffuser-section louver. Still another shape may be obtained by inverting these louvers forming a nozzle section louver.

By considering the various shapes, the five shape dimensions, and other variables such as density, viscosity, and velocity of the air it is possible to correlate by the use of dimensionless groups the characteristics of the louvers. For instance, the width coefficient, which is a ratio of the width of a board to the distance between boards indicates the significance of the width of a louver board on louver performance. As the width coefficient increases, a decrease in the resistance of the louver occurs until a certain optimum width coefficient is obtained. Then a further increase causes an increase in the resistance. This may be explained by the fact that the increase in width causes a pressure recovery within the louver by improving flow conditions, but friction losses due to increased width then overcome this effect, and a net increase in the resistance results.

The question was asked whether Mr. Cobb's data would apply to a louver such as a venetian blind. I would say no, because the data presented is not general enough, that is, it does not consider all of the significant variables.

The question was also raised about the practicality of the data from the standpoint of entrance losses to the louver. Mr. Cobb's results are based on a section within an infinite louver, and hence such losses are not considered. I am now preparing a paper on this problem, and I hope to obtain a solution through the use of an entrance contraction coefficient and an exit expansion coefficient.

AUTHOR'S CLOSURE: The author is very grateful for the interesting and stimulating comments made on this paper. As stated in the introduction to the paper, this was the first of a series of research programs being conducted by the Texas Engineering Experiment Station to evaluate the resistance of materials to air flow. C. W. Bevier continued the louver program by evaluating the effects of width and non-standard, cross sectional shapes. His comments on the effect of louver width were greatly appreciated. F. W. Rabe's discussion on the difference in entrance conditions for

* Student at Texas A. & M. College.

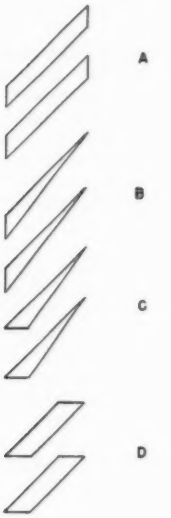


FIG. A. GEOMETRY OF THE LOUVER

A. Standard Louver
B. Diffuser—Section Louver
C. Nozzle—Section Louver
D. Inverted Louver

the air flow to air intake-louvers in a counterflow, induced draft cooling tower and those louvers tested for the paper should prove helpful in applying the experimental results.

In answer to R. W. McKinley's questions on how the data could be applied to such a cross section as that of a venetian blind, it is possible that geometric similarity could exist for some of those louvers tested with a free area ratio near unity. The effect of curvature will have to be evaluated experimentally. To what limits the results presented might be extended, a question raised by Dr. Nottage, is not readily determinable without additional experimentation. The independence of $\frac{P}{\rho V^2}$ with $\frac{RV\rho}{\mu}$ for free area ratios greater than 0.5 would be expected for higher values of $\frac{RV\rho}{\mu}$ than those tested for, but not for all values. Dimensionless groups were used for correlation and ease of plotting.

Dr. Gilman's discussion was especially appreciated and the analysis of the data by Dr. Gilman will prove invaluable in determining the worth of the paper. The answer to his question: "Is this really a Reynolds number $\left(\text{the group } \frac{RV\rho}{\mu}\right)$ at all in the true sense of the definition?" is no for the reason pointed out in Dr. Gilman's discussion. There is no mention of Reynolds number in the paper and such mention was carefully avoided.

The answer to his second question "What particular significance does the author attach to such a definition of Reynolds number?" will also clarify the author's position on the question of the difference in importance between the inlet duct velocity and the velocity of the air flow inside the louver. Major significance is attached to inlet duct velocity because of the practicability of measurement.

1497

No. 1497

CIRCUIT ANALYSIS APPLIED TO LOAD ESTIMATING

By H. B. NOTTAGE*, SANTA MONICA, CALIF. AND G. V. PARMELEE**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

AIR CONDITIONING engineers have long recognized that the capacity of a structure for storing heat has a significant effect on the equipment load when the structure is subjected, as in summer, to a variation in solar radiation intensity and outdoor temperature. Recent years have seen the development of means of computing rates of heat flow through walls and roofs subjected to steady periodic variations in the weather elements. These take into account the heat capacity of only the outer sections of the building. Heat gain data for sunlit glass sections of various types also are now available, and convenient tables for establishing instantaneous heat gain for walls, roofs and glass have been published in THE GUIDE for some years. However, these data give instantaneous rates of heat entry into the air conditioned space, and therefore account must be taken of the fact that the radiation component of these gains is not directly transferred to the air (except for a small amount of absorption) but instead warms the interior surfaces of the space. Because these interior sections absorb and store part of this radiant energy, re-radiate part, and transfer the balance by convection to the room air, the actual load on the conditioning system is considerably different from the instantaneous rate of heat flow into the space.

It is apparent that load estimating methods must be extended to include the heat capacity of the internal structure; that is, the entire thermal system must be considered. Improved knowledge of the time-variable behavior of the system load on an air conditioning installation will result in significant cost reduction by: (1) avoiding excessive installed capacity, and (2) operating with minimum power consumption throughout the daily operating cycle. Improved knowledge also will enable engineers to make more effective designs with full consideration of solar energy and of structural heat-storage capacity and will improve their understanding of how the building and its air conditioning system will adjust to weather changes.

In addition, the proper application of automatic controls will be enhanced by better knowledge of the significant instantaneous temperatures and heat-flow rates throughout an air-conditioned structure.

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Presented at the 60th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Houston, Tex., January 1954.

During the past few years, material progress has been made in dealing with problems of internal heat storage. (See references summarized in THE GUIDE[1] in the Chapter on Cooling Load.) This paper is a first progress report on the representation of thermal systems by *thermal circuits* and on the application of electrical-circuit techniques to the analysis of thermal circuits and the calculation of air conditioning loads. The objectives are:

1. To introduce the use of thermal circuits in thermal system analysis and load estimating.

2. To calculate temperatures, heat flow rates and cooling loads for an illustrative thermal circuit by mathematical techniques used in electrical circuit analysis.

Advisory guidance has been given by the Technical Advisory Committees on Cooling Load† and on Heat Flow Through Glass.‡

THERMAL CIRCUIT PRINCIPLES

An engineer's first step in problem solution is usually to make a diagram of the problem so that he can more readily visualize the relationship between problem elements. A thermal circuit diagram is simply a picture of a thermal system, and, in the absence of standardized symbols for thermal elements, it is expedient to use electric symbols to represent the thermal elements, each of which has its counterpart in electrical circuits (See Table A-1, Appendix A).

In Fig. 1 the upper diagram (a) shows a thermal system consisting of a wall exposed to the sun and outdoor air on one side and to room air and a cooling panel on the inside surface. Temperatures throughout the system are designated by numbers. Temperatures 3 and 4 are maintained at constant levels by appropriate mechanical systems which discharge the heat they collect from the circulating media to some heat sink, such as the outdoor air. A sink is simply an energy receiver that maintains a constant temperature regardless of the energy it receives. The variation with time of the solar intensity I, and the temperature and velocity of the wind are known, and the variations of temperatures 1, 2, and 5 with time are to be found.

These temperatures can be found by writing a heat balance equation for each point and solving these equations simultaneously. For example, the temperature at point 1 is such that the heat conducted to it from within the wall equals the heat it loses by convection to the room air at temperature t_4 and in the net radiation exchange with the cooling panel at temperature t_3. This equality, simply a statement of the energy conservation law, holds for both steady and nonsteady heat flow. Similarly, the temperature at point 2 is such that the rate at which it absorbs solar energy equals the sum of the rates at which it loses heat to the outdoor air by convection and to the wall interior by conduction. And if the thermal capacity of the wall is placed at point 5, the temperature of this point is such that the difference between the rates of heat conduction from 2 to 5 and from 5 to point 1 equals the rate at which heat is stored in the wall.

The relationship stated by these heat balances can be more clearly understood if the thermal system is pictured as a circuit shown in diagram (b) of Fig. 1.

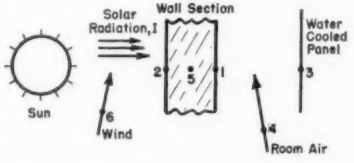
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[1] Exponent numerals refer to References.

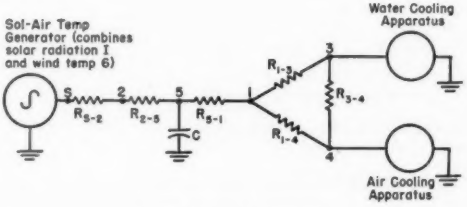
†*Personnel:* C. O. Mackey, *Chairman;* K. O. Alexander, John Everetts, Jr., L. F. Fisher, W. F. Friend, H. T. Gilkey, R. H. Heilman, H. W. Heisterkamp, A. J. Hess, R. C. Jordan, A. T. Jorn, C. F. Kayan, J. N. Livermore, P. E. McNall, Jr., I. A. Naman, H. B. Nottage, R. W. Roose, J. P. Stewart, H. B. Vincent, W. E. Zieber.

‡ *Personnel:* R. A. Miller, *Chairman;* N. B. Hutcheon, W. J. Arner, F. L. Bishop, E. W. Conover, D. D'Eustachio, W. B. Ewing, J. E. Frazier, J. S. Herbert, R. W. McKinley, E. C. Miles, D. R. Muir, H. B. Vincent.

It is commonplace now in air conditioning calculations to replace the two energy sources, the sun and the outdoor air, with a single source which produces a variable temperature, known as the sol-air temperature. This source is shown by the symbol of an electric generator, and the sol-air temperature, s, is connected to point 2 through a convection resistance, $R_{s\text{-}2}$. Similarly, conduction, convection, and radiation resistances connect the other junction temperature points in the circuit. A capacitor branch at point 5 represents the capacitance of the wall. Points 3 and 4 are tied together through a convection resistance, and are each connected to cooling equipment designated by circles in lieu of giving the complete thermal circuit of each apparatus. In practice, the equip-



a. THE THERMAL SYSTEM



b. THE THERMAL CIRCUIT

FIG. 1. THERMAL CIRCUIT FOR A SIMPLE THERMAL SYSTEM

ment is selected to have sufficient capacity to maintain points 3 and 4 at the desired constant levels. The equipment, the capacitor C, and the generator are connected to a common ground, which can be thought of as an energy sink of constant temperature. Principles of evaluating circuit component quantities and of selecting the number of capacitances for proper representation are discussed in Appendices A, B, and C. Numerical details for the illustrative problems are given in Appendix D.

The circuit resembles an electrical circuit only because electrical symbols have been used, but all quantities are thermal quantities. It should be realized that a thermal system is a network of energy-transfer resistances, capacitances and sources and, as such, can be represented by a circuit diagram.

Summarizing then, in a thermal network:

1. Thermal potentials are temperatures: units are in Fahrenheit degrees.
2. Thermal currents are heat flow rates; units are in Btu per hour.
3. Resistances transfer heat with a corresponding temperature drop; units are Fahrenheit degrees per Btu per hour.

4. Capacitances are heat storage elements; units are Btu per Fahrenheit degree.

5. Thermal sources produce heat, thermal sinks receive heat; units are Btu per hour.

Since the mathematical equations for periodic heat flow have counterparts in alternating current circuits, it is advantageous to use applicable methods of electrical network analysis that have already been well developed. This idea is not new, but the application of circuit diagrams and circuit analysis to thermal systems and control problems in air conditioning appears to have lagged behind the possibilities. Further discussion of circuit techniques is contained in memoranda** and presented to the Technical Advisory Committees.

Solving problems by the thermal circuit method requires the following data and steps:

1. Complete description of the physical system, involving: (*a*) dimensions; (*b*) thermal properties; (*c*) input cycles of temperature (potential) and/or heat flow rate (current); (*d*) independent variables in the surroundings of the physical system.

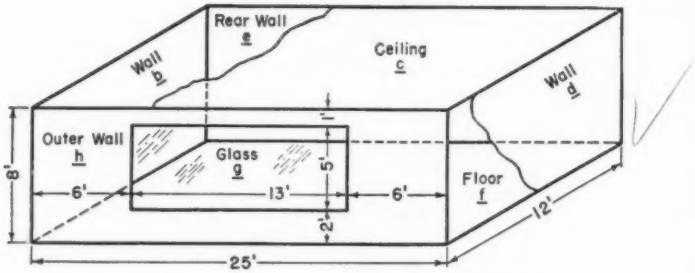


FIG. 2. SKETCH OF ROOM FOR ILLUSTRATIVE EXAMPLE

2. Calculation of resistances and capacitances.

3. Writing of circuit equations. These can be kept linear in the majority of problems.

4. Solving of circuit equations.

5. Practical judgment of results.

Functional details, as presented in the Appendices to this paper, may appear formidable to the uninitiated. In reality, all operations presented can be systemized and reduced to (1) tabular forms, and (2) sets of instructions. No higher mathematics is necessary. Problems are solved by the application of (1) arithmetic, (2) vector representation of periodic phenomena, and (3) algebra, for the solution of simultaneous equations.

Difficulties can arise in complicated circuits. For example:

1. There may be so many simultaneous equations to solve that the sheer labor is imposing. This can be conquered, however, through the use of standard calculating machines.

2. If resistances and capacitances vary with potentials or currents, then the equations become non-linear and are much more difficult to solve. This generally calls for a special analogue.

** Copies are available to A.S.H.V.E. Members upon request to the Research Laboratory.

PROCEDURES IN ANALYSES

Equations and tables are involved here. Study is required for proper understanding. Details have been placed in Appendices.

The presentation at this point will be limited to an illustrative example. For orientation purposes, however, the titles of the Appendices are listed as follows:

A. Definition and Inter-Conversion of Corresponding Thermal and Electrical Quantities, With Vector Notation.

B. Writing Circuit Equations.

C. Lumping of Circuit Parameters.

D. Calculation Details and Results for Thermal Resistances and Capacitances.

E. Circuit Input Potentials, Equations to be Solved, and Solutions Obtained.

F. Periodic Heat Gain by the Method in THE GUIDE for the Illustrative Example.

G. Treatment of Directly Transmitted Radiation and Radiant Energy Absorption by Room Air.

ILLUSTRATIVE EXAMPLE

A step-by-step summary will demonstrate the procedure. Two cases are to be considered, namely:

Case I Room with single glass window.

Case II Room with glass block window.

The problem has further been subdivided into the following two phases.

Phase A Analysis *excluding* the radiation directly transmitted through the glass.

Phase B Additional effects of radiant energy directly transmitted through the glass.

The present paper treats Cases I and II, *Phase A only*. Successive steps to the desired results are given in following paragraphs A to H.

A. Sketch the physical system and list the restrictions imposed. Fig. 2 shows the chosen room schematically. Table 1 describes the material comprising the room components and lists the thermo-physical properties. Postulates are:

1. Cooling-season conditions prevail.

2. The room has one outside wall with due-north exposure.

3. The room is surrounded by similar rooms, such that the floor, ceiling, and interior partitions contain planes of no heat flow. These planes are assumed to be at the respective mid-sections.

4. All surfaces are isothermal planes—room, outside wall, and glass.

5. Room air is maintained at a uniform constant temperature.

6. No heat sources or furnishings are in the room.

7. No moisture transfer is involved.

8. No radiant energy is absorbed by the room air.

9. Independent variables, sol-air temperature, and corresponding indoor-surface glass temperature, cycle in a steady periodic state.

10. Infiltration is regarded as an instantaneous superposed load.

11. All surfaces radiate diffusely.

12. Thermal resistances and capacitances are independent of potentials or currents. For a discussion of the lumping of capacitances, see Appendix C.

B. Draw the thermal circuit.

1. Fig. 3 gives the circuit wherein the symbols are as described in the following paragraphs. (Also see Table A-1, Appendix A.) The dotted branches refer to Phase B of the problem, which is to be treated in a subsequent paper.

2. Circuit junction points represent places where temperatures are to be determined; the letter symbols refer to the corresponding indoor surfaces of Fig. 2. These junction points include all room surfaces, the room air, a, and points k and m within the outer wall at positions determined by principles and calculations outlined in Appendices C and D. The constant room air temperature, at a, is taken as the datum.

3. Thermal currents flow in the various circuit branches; units are Btu per hour. In this paper current flow in a branch equals the product of potential (temperature) difference between the junction points of the branch times the admittance, Y of the branch. This term is explained in detail in Appendices A and B. For the present, Y can be regarded as comparable to conductance in steady-state heat flow.

4. Room-air junction, a, is grounded through the system-load meter, which measures the net thermal current required to hold the air at constant potential (temperature). This gives the instantaneous load on the air conditioning equipment. In the circuit analysis, the *ground* is a so-called sink which can receive current of any magnitude and always remain at constant potential, and this is the manner in which the constant room air temperature is represented. Transposed to the air-conditioning system, the ground symbolizes equipment which somehow will always maintain the room air at

TABLE 1—ASSUMED STRUCTURAL DETAILS AND THERMAL PROPERTIES FOR THE ILLUSTRATIVE EXAMPLE

Room Component	Designation in Fig. 2	Material	Total Thick. in.	Surface Area sq ft	k[a] Btu/(hr) (sq ft) (F deg/ft)	ρ lb/cu ft	c_p Btu/lb F deg	a sq ft/hr
Floor	f	Concrete, no covering	4	300	0.41	97	0.21	0.020
Ceiling	c	Same as floor						
Interior Partitions	e	Gypsum lath & plaster	⅞	200	0.275	70	0.23	0.017
		Wood studs, 1¾ x 3¾ in. spaced 16-in. centers	—	—	0.085	30	0.40	0.007
	b, d	Same		96				
Exterior Wall	h	Brick	4	135	0.51[b]	112[b]	0.20	0.023
		Mortar	½	135	1.0	115	0.22	0.039
		Cinder-concrete blocks	8	135	0.40[b]	57[b]	0.21	0.033
		Furring space, lath & plaster, taken to be same as for interior partitions.						
Glass	g	See Fig. 5	Case I—Single common window glass, 65 sq ft Case II—Type I 8-in. glass blocks Smooth exterior faces, wide ribs or flutes on interior faces, 65 sq ft					

[a] Symbols: k = thermal conductivity
ρ = density
c_p = unit heat capacity at constant pressure
a = thermal diffusivity

[b] For solid substance with no core holes, having the same net weight and over-all thermal resistance as the actual structural units.

a constant temperature, no matter what the instantaneous load may be. This latter concept is an idealization which could be improved by placing equipment components in the circuit, but this is beyond the present scope.

5. Room-radiation current branches run directly between the surfaces concerned; each surface is connected to each of the other surfaces which it *sees* by branches with radiation resistances R_{be}, R_{bc}, etc. The resistance values are determined in accordance with Appendix D.

6. Convection-current branches run directly between the surface concerned and the room air, a. The resistances of these branches, R_{ba}, R_{fa}, etc. are determined in accordance with Appendix D.

7. In accordance with postulate 3 of paragraph A, the circuit branches between the surfaces of the inner walls, floor, and ceiling and the mid-points of these sections

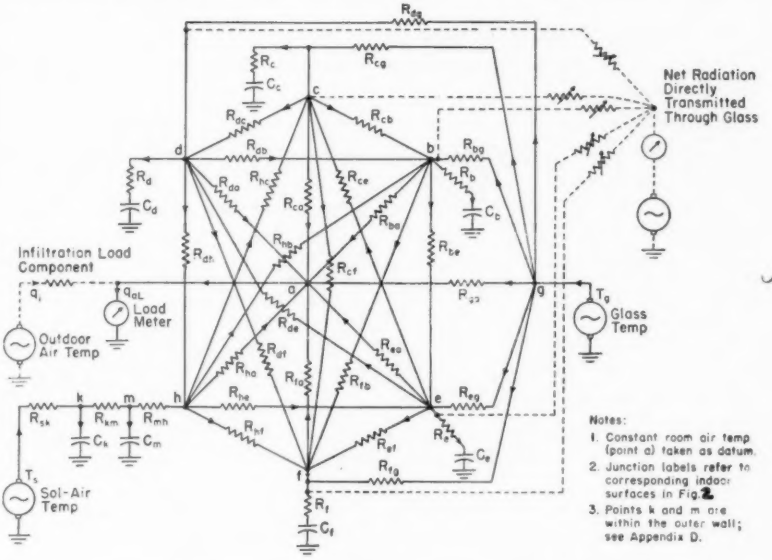


FIG. 3. THERMAL CIRCUIT TO BE SOLVED (DOTTED PORTIONS NOT INCLUDED IN THE ANALYSIS PRESENTED IN THIS PAPER)

consist of a conduction-current resistance, such as R_f, in series with a structural capacitance, such as C_f, connected to ground. A thermal current flows into or out from this capacitance (heat storage element) only through the resistance, R_f, in accord with the difference in potential between circuit points f and C_f. Resistance and capacitance values are determined as in Appendices C and D.

8. The conduction-current branch between the sol-air temperature and the indoor surface, h, of the outer wall consists of a series of resistances R_{sk}, R_{km}, and R_{mh} with structural capacitances C_k and C_m tied into this branch at points determined by methods outlined in Appendices C and D. Current flows into or out from these capacitances in accordance with potential (temperature) differences.

9. Independent input variables are the sol-air temperature, T_s, and the corresponding indoor glass-surface temperature, T_g. These sources are depicted as generators which

generate potentials (temperatures) which vary with time in accordance with the conditions of the problem (see Figs. 4 and 5).

10. Infiltration and directly-transmitted radiation are drawn dotted as superposed components, with analysis reserved for the paper covering treatment of Phase B.

C. Express the cyclic input variables in the Fourier series.

1. This requires (a) a cycle shown graphically, and (b) a standard calculation form and arithmetic.

2. For the present problem, the sol-air temperature[2] is shown graphically in Fig. 4. The inner-surface glass temperatures[3] are shown in Fig. 5.

3. If the required series is lengthy, a harmonic analyzer will be helpful.

4. Data and results are summarized in Appendix E.

D. Write the circuit equations.

1. These can be written systematically, guided by a form sheet and a set of instructions if advisable.

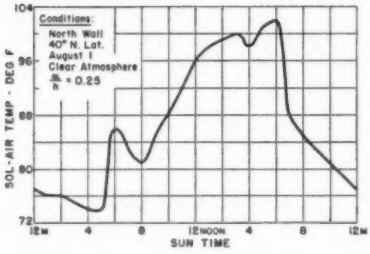


FIG. 4. SOL-AIR TEMPERATURE CYCLE

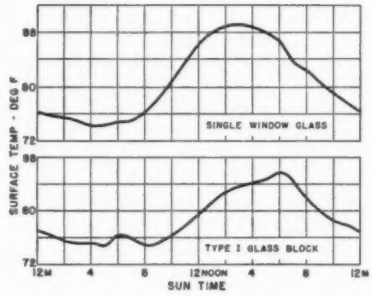


FIG. 5 *(right)*. INDOOR GLASS SURFACE TEMPERATURES COMPATIBLE WITH SOL-AIR TEMPERATURES OF FIG. 4

2. It is convenient to treat the potentials as unknowns. There will be one equation for each unknown junction-point potential (temperature).

3. Results are summarized in Appendix E.

E. Calculate the circuit resistances and capacitances.

1. Methods and results are summarized in Appendix E.
Appendix A.

2. Resistances and capacitances are combined to give admittances as described in

F. Solve the equations for the circuit potentials (temperatures).

1. Possible methods are: (a) algebraic, by substitution or determinants; (b) numerical; or (c) computing machine.

2. A computing machine was employed in the present instance. Results are summarized in Appendix E.

3. In complicated cases of more equations than can be handled by a conventional computing machine, or with non-linear equations, an analogue will be helpful or may be the only workable method.

G. Calculate the thermal currents.

1. Methods and results are summarized in Appendix E.

2. Check the solution by making circuit balances at each of the circuit junction points.

H. Determine the instantaneous load and appraise the results.

This is discussed in the following sections. The load is the instantaneous sum of the currents at point a, Fig. 3.

RESULTS

All results include Cases I and II, namely, single window glass and glass block respectively, but only Phase A. Phase B, which includes the load due to

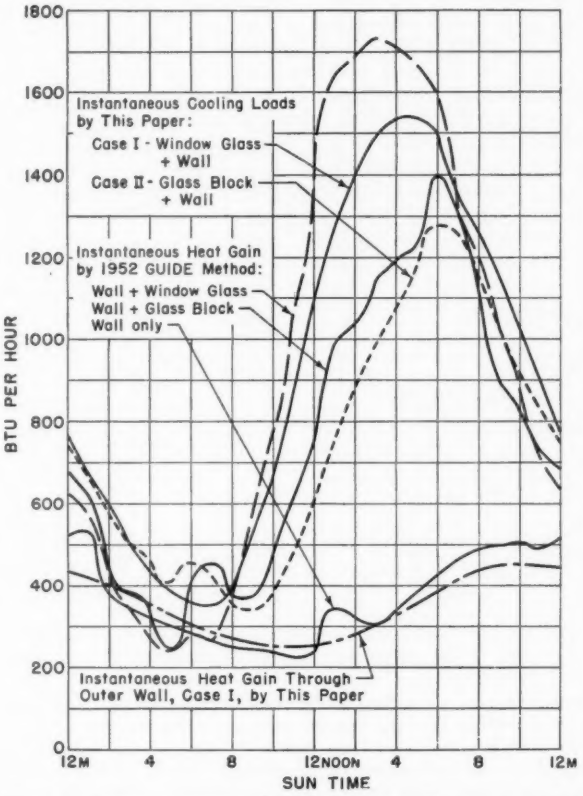


FIG. 6. INSTANTANEOUS COOLING LOADS COMPARED WITH INSTANTANEOUS HEAT GAINS. DIRECTLY TRANSMITTED RADIATION FROM SUN AND SKY IS NOT INCLUDED

directly transmitted solar radiation, is not part of the present results, but remains to be covered in a subsequent paper.

1. *Instantaneous system load:* Fig. 6 compares the thermal-circuit calculated system load with the heat gains through the outer wall and glass as determined by THE GUIDE, 1952 method.[1] (Inclusion of directly-transmitted radiation would increase the spread between the respective peaks). The fact that instantaneous heat gains overestimate the peak system load has been pointed out before[1, 4].

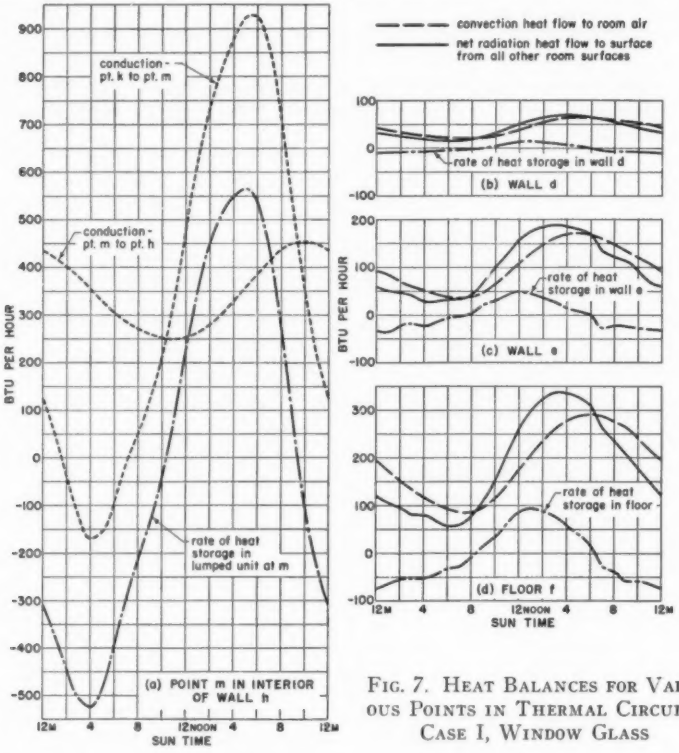


FIG. 7. HEAT BALANCES FOR VARIOUS POINTS IN THERMAL CIRCUIT: CASE I, WINDOW GLASS

The fact that the night-time instantaneous system load exceeds the instantaneous heat gain rate is proper. In the steady state, the net 24-hr heat gain must equal the total 24-hr system heat removal.

2. *Instantaneous heat gain through outer wall:* Fig. 6 also compares the two methods of calculating the instantaneous heat gain through the outer wall. The curve for the wall only for Case II is nearly the same as for Case I and therefore is not shown. Both methods contain approximations and the comparison shows order-of-magnitude agreement. The method of THE GUIDE is approximate because it consists of a steady state heat flow with a superimposed heat flow curve that exactly follows the sol-air temperature curve except that it is displaced by the time lag of the wall. The amplitude of this curve is less than that of the sol-air temperature

curve, but the approximate method does not take into account the greater damping effect imposed by the wall capacitance on the higher frequency *bumps* in the sol-air temperature curve (see Fig. 4). The circuit method gives a relatively smooth wave. Investigation showed that the current flow between points m and h, that is, the rate of instantaneous heat flow into the room from the outer wall, gave results for the first harmonic within 3 percent of the flow rates based upon four harmonics. It seems doubtful that inclusion of harmonics of higher order than the 4th would significantly alter the results obtained for Phase A of this illustrative problem. The question of

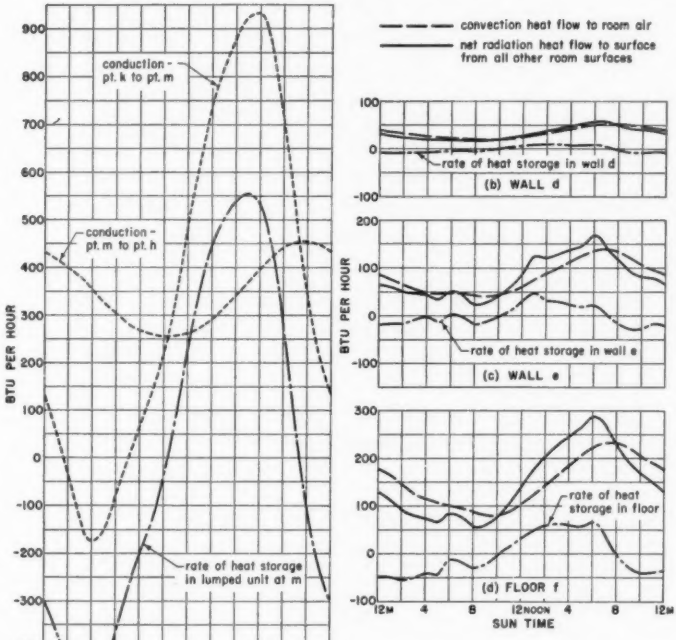


FIG. 8. HEAT BALANCES FOR VARIOUS POINTS IN THERMAL CIRCUIT: CASE II, GLASS BLOCK

further subdividing the outer wall in the circuit method is concerned with a type of approximation which is considered in Appendix E.

3. *Component thermal currents:* Figs. 7 and 8, for Cases I and II respectively, show thermal currents of interest throughout the room. In each instance, the storage rate is the difference between the heat flow rate into a circuit point by conduction, convection, or radiation or any combination of them and the heat flow away from that point by the same modes. Over the 24-hr period the stored heat is zero. The following may be noted. (Refer to the room sketch of Fig. 2.)

Figs. 7a and 8a: The difference between the heat flow through the outer masonry-wall portion of the circuit with the single window, and that through this wall with the

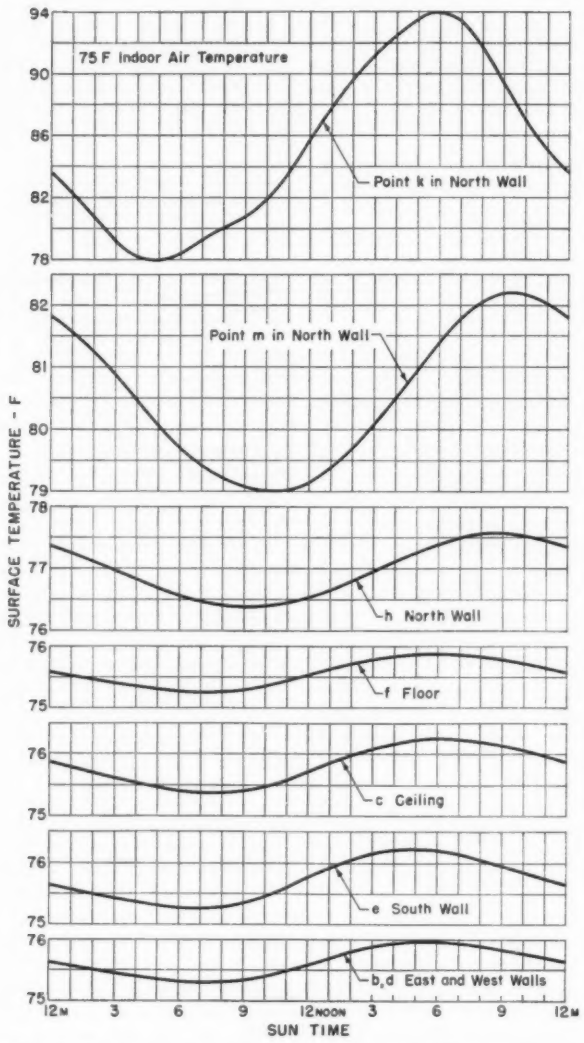


FIG. 9 TEMPERATURES IN OUTER WALL AND TEMPERATURES OF INDOOR ROOM SURFACES: CASE I—FLAT GLASS WINDOW

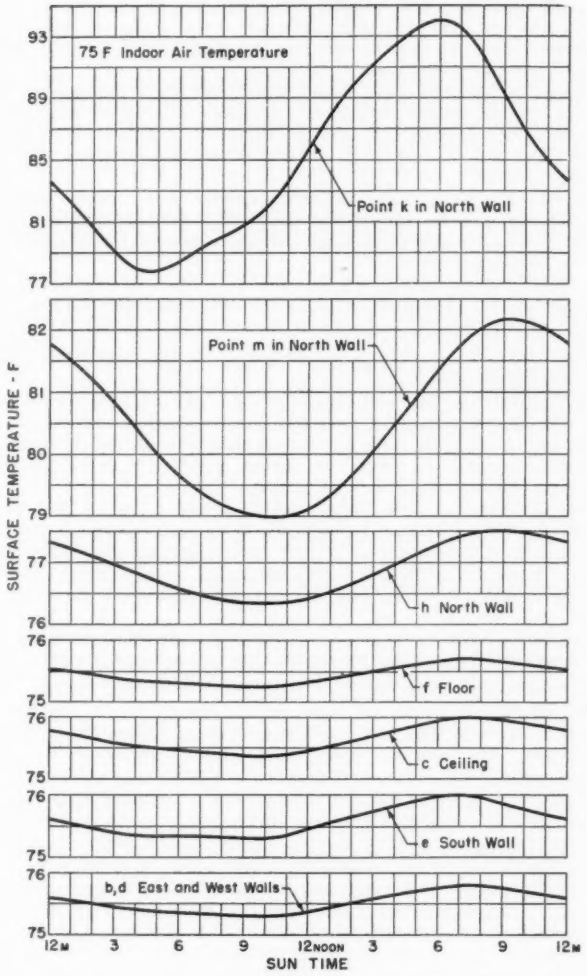


FIG. 10. TEMPERATURES IN OUTER WALL AND TEMPERATURES OF INDOOR ROOM SURFACES: CASE II—GLASS BLOCK WINDOW

glass block window is negligibly small. This is logical because the glass and the outer wall are parallel heat flow paths in each case, linked only by convection from the same air currents plus radiation reflection within the room. Also, the wall was of very heavy construction in this instance, and this helped greatly to make its behavior insensitive to differences in the glass portion.

Figs. 7b, 8b: For the end walls, d and b, the glass block produces an appreciably greater time lag and a slight decrease in amplitude. This agrees with expectations.

Figs. 7c, 8c: The rear wall, e, is strongly influenced by radiation from the glass and outer wall. The greater resistance and capacitance of the glass block have the expected effect.

Figs. 7d, 8d: The floor, f, has a high capacitance and shows a very large time lag in combination with the glass block. The ceiling would have a similar behavior. While this effect individually is large, its final influence upon the system load depends upon the entire system circuit.

The breakdown of the system behavior into component effects, which is afforded by the circuit method, gives a flexible basis for design analyses. In this way, causes, effects, and variables may be isolated and studied in their proper light.

4. *System temperatures:* Figs. 9 and 10 summarize the temperature cycles at the important points in the circuit.

Appreciable differences between glass block and window glass would be expected where the radiation currents were strongest. The results show this for the floor, ceiling, and interior south wall. Still, the absolute magnitude of the differences is small. Moreover, the amplitudes of the interior-surface temperature cycles also are small—small enough to be neglected for some purposes.

These findings pertain only to Phase A of the problem. In the illustrative system, the maintaining of a constant air temperature, along with moderately low convection resistances, serves to hold down the interior-surface temperature amplitudes. Including the directly transmitted radiation will change this, perhaps slightly, for a northern exposure. With a southern exposure, much greater orders of magnitude would be expected.

The comparative times for the peaks of the temperature curves show the interesting effect of radiation and convection circuit branches operating in parallel, for example:

a. The interior south wall, of light construction and directly opposite the window, reaches its peak first.

b. The east and west walls come next in time phase, followed very closely by the floor and ceiling. The floor and ceiling receive more radiation than the end walls, but their capacitance is greater, and this is important in the time lag.

c. The circuit point m, in the middle of the north wall resistance, has the latest peak of all. This is a consequence of the room surfaces *seen* by h being warmed by radiation from the glass several hours before the peak of the temperature wave traveling through the outer wall reaches the inside surface, h. The exchanges of radiation between h and these room surfaces require that h rise in temperature in accordance with the rise in temperature of the other room surfaces, but with a time lag. It will be noted that the temperature at h peaks later in Case II than in Case I. This agrees with the fact that the temperature curve for glass block peaks later than that for single window glass. (See Fig. 5.) A lighter outer wall construction would change the phase relationship of the inner surface temperatures.

DISCUSSION

Useful answers to problems of the type illustrated are needed with a minimum of time and cost. All methods of analysis begin with a definition of the physical system which establishes the thermal resistances, capacitances and sources in-

volved. Possible means of obtaining results include (1) differential equations, (2) numerical methods[5], (3) analogues, (4) circuit calculations, and (5) full-scale experiments.

Comments on these various means follow.

Differential equations are a general and fundamental method, capable of handling transient and aperiodic conditions. If the equations remain linear, and if there are not too many of them to be solved simultaneously, anyone with the necessary mathematical acquaintance can achieve results. This conventional attack becomes unwieldy, however, when confronted with even the relatively simple problem illustrated in this paper.

Numerical methods really are a variation of the circuit approach, wherein the simultaneous equations are solved by arithmetic[5]. This is often the best technique when but a single problem is to be solved.

Analogues are convenient representations of circuits, wherein answers for time-variable cases can be obtained for quite complicated systems, including non-linear behavior, by laboratory experiments. The deterrent here is the cost of the analogue equipment, which can be justified only by a strong need for extensive use.

Circuit calculations are very convenient for either steady or steady-periodic systems. Advanced analytical techniques, beyond the simple method of this paper, offer extremely powerful and general tools.[6] For the contemplated problems of air-conditioning interest, the method of calculating-machine solutions of circuit equations offers high promise. This can be handled in the ordinary engineering office by outside hiring of computing services, if necessary.

Full-scale experiments are practicable in principle, but they are exceedingly difficult to realize except under controlled laboratory conditions. These also are very time-consuming and present a very lengthy and expensive program if all variables are to be studied over their complete ranges.

Above all, no problem solutions can be better than the physical data upon which they are based.

Present needs are believed to comprise:

1. Improved data on thermal resistances in full-scale rooms. This need is being met largely by current research in the ASHVE Environment Laboratory.

2. Standardized data on design weather conditions, such that circuit inputs can be readily established for all parts of the country. This is one of the goals of ASHVE research.

3. Extension of the glass design data, such that standardized input cyles can be established for all parts of the country and for varied indoor conditions. This can be done with information presently available at the ASHVE Research Laboratory.

4. A careful study of results attainable, costs, time, and practical-application values for the different methods of solving periodic-condition design and analysis problems. Present judgment is that a combination of circuit analysis and an analogue (for the more complicated cases) will be superior for broad and flexible use. This, however, requires further substantiation. An ASHVE cooperative project now under way at the University of California, Los Angeles, is expected to provide valuable guidance.

5. Continuing educational development of circuit methods. This includes recognizing (1) the type of design data which will ultimately be needed for automatic controls, and (2) the time-variable response of equipment.

RECOMMENDATIONS

1. Persons interested in the details of circuit methods should study the Appendices.

2. Phase B and the final conclusion of the illustrative problem should be awaited before attempting complete quantitative judgment of the present results.

3. Extension of circuit principles into problems of automatic control is an inviting prospect which should be developed.

CONCLUSIONS

1. Thermal circuit principles and methods of analysis have been demonstrated to yield practical results in the analysis of periodic heat transfer rates and temperatures in air-conditioning systems.

2. The difference between the instantaneous rate of heat gain, by the method used in THE GUIDE, and the instantaneous system load by circuit analysis, (Fig. 6) is believed to demonstrate the practical value and economic savings which can be gained by designing according to circuit principles.

REFERENCES

1. HEATING, VENTILATING, AIR CONDITIONING GUIDE 1952, Chapter 12—Cooling Load. (Published annually by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y.).

2. THE GUIDE 1952, Chapter 12—Cooling Load, Table 9, p. 273.

3. These data are based upon THE GUIDE 1952, as follows: (*a*) single glass. Table 17, page 291, and (*b*) glass block designated as Type I for Table 24, p. 295. In both instances, the indoor temperature is 75 F with interior surfaces having an emissivity of unity, and the outer surface conductance is 4.0 Btu/(hr) (sq ft) (F deg). Slight cyclic departures from 75 F, or emissivities which are less than 1.0 but still high enough to neglect reflections, are not considered to be important deviations for present purposes.

4. Cooling Load from Sunlit Glass, by C. O. Mackey and N. R. Gay (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 321).

5. *Numerical Analysis of Heat Flow*, by G. M. Dusinberre (McGraw-Hill Book Co., Inc., New York 1949).

6. *Matrix Analysis of Electric Networks*, by P. LeCorbeiller (Harvard University Press, Cambridge, 1950).

APPENDIX A

Definition and Interconversion of Corresponding Thermal and Electrical Quantities, with Vector Notation

A specific illustration of the basis for quantitatively interrelating analogous variables in different physical systems will be given. Thermal and electric equivalents have been chosen for this purpose, but the method applies as well to other cases. Table A-1 gives the fundamental definitions. The vector notation is further explained in the text which follows.

$$N_t = \frac{\text{Magnitude of time unit in thermal circuit}}{\text{Magnitude of time unit in electrical circuit}} \quad \text{(A-1)}$$

$$N_Q = \frac{\text{Magnitude of quantity unit in thermal circuit}}{\text{Magnitude of quantity unit in electrical circuit}} \quad \text{(A-2)}$$

$$N_q = \frac{\text{Magnitude of current unit in thermal circuit}}{\text{Magnitude of current unit in electrical circuit}} \quad \text{(A-3)}$$

$$N_P = \frac{\text{Magnitude of potential unit in the thermal circuit}}{\text{Magnitude of potential unit in the electrical circuit}} \quad \text{(A-4)}$$

$$N_R = \frac{\text{Magnitude of resistance unit in thermal circuit}}{\text{Magnitude of resistance unit in electrical circuit}} \quad \text{(A-5)}$$

$$N_C = \frac{\text{Magnitude of capacitance unit in thermal circuit}}{\text{Magnitude of capacitance unit in electrical circuit}} \quad \text{(A-6)}$$

Following the above six definitions, there are three general relationships which must hold:

1. Defining equation for quantity,

$$N_Q = N_q N_t \quad \text{(A-7)}$$

2. Ohm's Law,

$$N_P = N_q N_R \quad \text{(A-8)}$$

3. Equivalence of time constants for resistance-capacitance circuits,

$$N_t = N_R N_C \quad \text{(A-9)}$$

Thus, there are three equations and six variables, and up to three of the variables may be assigned units as desired. These three chosen variables, however, must not occur all in one equation, because the three equations do not each contain all of the variables. Particular caution must be taken in observing that convention has established the *second* as the principal time unit in electrical terminology, whereas the *hour* is customary in thermal terminology.

Suppose, for example, that the following choices are made:

$N_C = 10^6$, or 1 microfarad = 1 Btu/F deg
$N_R = 10^{-6}$, or 1 megohm = 1 F deg/(Btu/hr)
$N_P = 1$, or 1 volt = 1 F deg

Then,

$N_t = N_R N_C = 1$, or 1 sec (electrical) = 1 hr (thermal)
$N_q = N_p / N_R = 10^6$, or 1 Btu/hr = 1 microampere
$N_Q = N_q N_t = 10^6$, or 10^6 Btu = 1 coulomb

In a practical case, the choice of interconversions should follow economy, experimental convenience and attainment of the desired experimental accuracy with equipment which may be on hand.

Vector notation is an essential part of the circuit method as developed here. This replaces the calculus and leads to algebraic methods instead of differential equations. Reference to any good text on alternating-current circuits is advised in this connection.

For any circuit branch, the potential and the current are related in terms of the impedance, defined as

$$\text{Impedance} = \text{Potential difference} \div \text{Current} \quad \text{(A-10)}$$

Symbolically,

$$Z = E/I \quad \text{(A-11)}$$

The vector impedance of a circuit branch containing resistance and capacitance in series is

$$Z = R - j X_c, \text{ F deg/(Btu/hr)} \quad \text{(A-12)}$$

where j is the operator $\sqrt{-1}$.

Another useful quantity is the admittance, defined as the reciprocal of the impedance, or

$$Y = 1/Z = I/E \quad \text{(A-13)}$$

Admittance is employed in this paper because of its convenience in giving the current in the form

$$I = Y E \quad \text{(A-14)}$$

The corresponding thermal notation is

$$q = Y (\Delta T) \quad \text{(A-15)}$$

The vector form of a cyclic potential is established from the fact that periodic functions are expressible by a Fourier series, of the form

$$T = a_0 + \sum_{n=1}^{\infty} (a_n \cos n\omega t + b_n \sin n\omega t), \text{ F deg} \quad \text{(A-16)}$$

where

a_0 = constant
a_n, b_n = Fourier coefficients
ωt = time angle, radians
n = order of harmonic

The corresponding vector notation, in accord with the sign convention of equation A-12, is

$$T = a_0 + \sum_{n=1}^{\infty} (a_n - jb_n), \text{ F deg} \quad \text{(A-17)}$$

The coefficients a_n and b_n can be established by arithmetical processes by following standardized calculation forms.

Table A-1 summarizes the vector notations needed in this paper.

ITEM	ELECTRICAL		THERMAL	
A. DEFINITIONS Quantity	Symbol	Units	Symbol	Units
Time Frequency	t f	sec cycles/sec	t f	hr cycles/hr
Charge or quantity	Q	coulombs	Q	Btu
Potential, in general Potential difference Datum potential	V V V_0, commonly zero	volts volts	T ΔT T_0	deg F deg F
Current	I	amperes = coulombs / sec	q	Btu/hr
Resistance	R	ohms = volts potential diff / amperes	R	(deg F)/(Btu/hr)
Capacitance	C	farads = coulombs charge / volts potential diff	C	Btu/deg F
B. SYMBOLS				
Source of constant potential				
Source of periodically-varying potential				
Sink or "ground"				
Resistor				
Capacitor				

TABLE A-1—CORRESPONDING THERMAL AND ELECTRICAL QUANTITIES AND RELATIONSHIPS

(Table concluded on page 77)

APPENDIX B

Writing Circuit Equations

Circuit equations are written in accord with Kirchoff's laws for electrical networks. These state that: (1) the currents must balance at any junction point, whereby inflow equals outflow, and (2) the potential differences must balance in going around any closed mesh.

The current-balance method is most convenient for circuits of the present type. In this, the current in any branch is written in terms of the admittance and the potential difference across the branch as

$$q = Y(\Delta T), \text{ Btu per hour} \quad \text{(B-1)}$$

Refer to Fig. B-1 for a current balance at some general junction point x, connected to n other points. Assume current directions as indicated by the arrows. An erroneous assumption will be corrected in the algebraic sign of the answer. Summing the instantaneous currents,

$$q_{1x} + q_{2x} + q_{3x} + \ldots + q_{nx} = q_{xo} \quad \text{(B-2)}$$

The subscript sequence matches the assumed direction of current flow.
For any branch n the current is

$$q_{nx} = Y_{nx}(T_n - T_x), \text{ Btu per hr} \quad \text{(B-3)}$$

where

Y = admittance, Btu/(hr) (F deg).
T = potential, F deg.

Substituting potentials according to equation B-3 into equation B-2,

$$Y_{1x}(T_1 - T_x) + Y_{2x}(T_2 - T_x) + Y_{3x}(T_3 - T_x) + \ldots\ldots + Y_{nx}(T_n - T_x) = Y_{xo}(T_x - T_o) \quad \text{(B-4)}$$

C. USEFUL CIRCUIT RELATIONSHIPS

Ohm's Law for resistance branch	$V = IR$	$\Delta T = qR$
Relation between capacitance, potential and current	$C = \frac{Q}{V} = \frac{\int_0^t I\,dt}{V}$ or, $C = \frac{I}{\left(\frac{dV}{dt}\right)}$	$C = \frac{Q}{T - T_0} = \frac{\int_0^T W c_p dT}{T - T_0} = \frac{\int_0^t q\,dt}{T - T_0}$ or, $C = \frac{q}{\left(\frac{dT}{dt}\right)}$
Capacitive reactance, defined	$X_c = \frac{1}{2\pi f C}$	$X_c = \frac{1}{2\pi f C}$
Vector notation, with current along real axis	$Z = -jX_c$	$Z = -jX_c$
Impedance for R-C series branch, defined	$Z = \frac{V}{I}$	$Z = \frac{\Delta T}{q}$
Magnitude	$\lvert Z\rvert = \sqrt{R^2 + (-X_c)^2}$	$\lvert Z\rvert = \sqrt{R^2 + (-X_c)^2}$
Time interval by which potential lags current	$\Delta t = \frac{1}{2\pi f} \tan^{-1}\left(\frac{X_c}{R}\right)$	$\Delta t = \frac{1}{2\pi f} \tan^{-1}\left(\frac{X_c}{R}\right)$
Vector notation	$Z = R - jX_c$	$Z = R - jX_c$
Admittance for resistance branch	$Y = \frac{1}{Z} = \frac{1}{R}$	$Y = \frac{1}{Z} = \frac{1}{R}$
Admittance for capacitance branch	$Y = \frac{1}{Z} = \frac{1}{X_c}$	$Y = \frac{1}{Z} = \frac{1}{X_c}$
Admittance for R-C series branch	$Y = \frac{1}{Z} = \frac{R}{R^2 + X_c^2} + \frac{jX_c}{R^2 + X_c^2}$	$Y = \frac{1}{Z} = \frac{R}{R^2 + X_c^2} + \frac{jX_c}{R^2 + X_c^2}$
Vector sketches		$\varphi = \tan^{-1}\left(-\frac{X_c}{R}\right)$

Combining terms,

$$Y_{1x}T_1 + Y_{2x}T_2 + Y_{3x}T_3 + \ldots\ldots + Y_{nx}T_n + Y_{xo}T_o - (Y_{1x} + Y_{2x} + Y_{3x} \ldots\ldots\ldots Y_{nx} + Y_{xo})T_x = 0 \quad \text{(B-5)}$$

Equation B-5 is the basic form applicable to each junction point in a circuit. These equations can be written directly by inspection for any junction point x in the form

$$\sum_{n=0}^{n=n} Y_{nx}T_n - \left(\sum_{n=0}^{n=n} Y_{nx}\right)T_x = 0 \quad \text{(B-6)}$$

There will be one equation for each junction point. The admittances are established from the given physical data, and the potentials are the unknowns. With one equation for each unknown junction-point potential, the problem can be solved. In the present method, all quantities enter in vector notation.

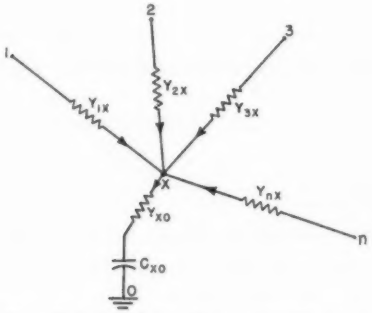


FIG. B-1. TYPICAL THERMAL-CIRCUIT JUNCTION POINT

APPENDIX C

Lumping of Circuit Parameters

The solution to any circuit is the answer to a specifically stated problem. How closely this solution may represent the behavior of some original physical system depends upon how well the circuit itself portrays the physical system.

The room of Fig. 2 in the paper is a complicated three-dimensional thermal system. The circuit of Fig. 3 is linear, with unidimensional flow paths; and this is not an exact portrayal of the room. How adequately the approximations involved may serve to yield engineering results is a matter of judgment and experience. Beyond the idealizations stated in the paper, a basic consideration is the lumping into discrete, point-concentrated units which has been imposed on the conduction-path capacitances.[C-1]

Consider, for instance, unidimensional heat conduction across a slab in the steady periodic state. For each small increment of distance along the heat-flow path, there will be an associated small element of thermal capacitance. The capacitance of the slab thus is said to be continuously distributed along the heat-conduction path.

The problem here is analogous to the long electrical alternating-current transmission line with resistance, capacitance, and inductance continuously distributed

along the line. This type of problem can be analyzed by a proper application of the calculus; but by lumping the distributed quantities into discrete, point-concentrated units, the problem can be solved with algebra and arithmetic, using vector notation.

The behavior of the lumped line will differ from the behavior of the continuously-distributed line. This difference, however, can be kept inconsequentially small by a *proper choice of the length of line to be lumped.*

For a resistance-capacitance line, it has been shown[C-2] that the calculated current for a lumped line will be within 5 percent of the calculated current for a continuously-distributed line if the length of conductor to be lumped is one-eighth of a wave length. With this increment of line length, the lumped capacitance is connected to ground at the mid-point, and the resistance is divided equally on either side of the grounded capacitance branch.

Applying this to periodic heat conduction, the wave length is

$$\lambda = 2\sqrt{\frac{\pi a}{f}}\text{, ft} \quad \ldots\ldots\ldots\ldots \text{(C-1)}$$

where

a = thermal diffusivity, square feet per hour.
f = frequency, cycles per hour.

Thus, the length of path to be lumped, l, is $(\lambda/8)$ or less, *i.e.*,

$$l \leq 0.44\sqrt{\frac{a}{f}}\text{, ft} \quad \ldots\ldots\ldots\ldots \text{(C-2)}$$

Note that f varies with the harmonic concerned, so that a compromise is usually advised when several harmonics are involved.

The outer wall of Fig. 2 is represented by the circuit portion between h and T_a in Fig. 3. Calculation and location of the lumped capacitances C_k and C_m are covered in Appendix D. The interior structure capacitances are lumped in single units and grounded as shown in Fig. 3.

REFERENCES

C-1. Accuracy of Measurements in Lumped R-C Circuits as Used in the Study of Heat Flow, by V. Paschkis and M. P. Heisler (*Transactions American Institute of Electrical Engineers,* Vol. 63, p. 165-171, April, 1944).

C-2. *Electromechanical Transducers and Wave Filters,* by W. P. Mason, (D. Van Nostrand Company, New York, 1942, First Edition, p. 62).

APPENDIX D

Calculation Details for Thermal Resistances and Capacitances

RADIATION RESISTANCES

1. *Equation employed:* Consider two absolute temperatures, T_1 and T_2. If $T_{av} = \frac{T_1+T_2}{2}$, a very good approximation is

$$T_1^4 - T_2^4 \cong (T_1 - T_2)(4\,T_{av}^3) \quad \ldots\ldots\ldots\ldots \text{(D-1)}$$

This introduces an error of only ten percent when $T_1 = 2T_2$, and therefore the relationship is quite satisfactory for most problems in air conditioning.

The thermal resistance for the exchange of *direct radiation* between two plane surfaces at temperatures T_1 and T_2 respectively, and based upon the area A_1, is

$$R_{(1\rightarrow 2)D} = \frac{1}{\sigma A_1 F_s F_e 4T_{av}^3} \quad \text{. . . . . (D-2)}$$

where

σ = radiation constant, Btu/(hour) (square foot) (Fahrenheit degree abs)^4.
A_1 = area of surface 1, square foot.
F_s = shape factor for use with A_1, dimensionless.
F_e = emissivity factor, dimensionless.

If the surface emissivities are high, say 0.9 or greater, then the influence of reflected radiation may be neglected.

Equation D-2 presumes that the surfaces are separated by a non-absorbing medium.

The radiation resistance depends upon T_{av}, and the average temperature cannot be fixed without knowing both T_1 and T_2. Hence, when surface temperature is unknown, an initial assumption is required for T_{av} which must be checked against the final solution and revised if necessary.

2. *Factors entering the resistance:* The constant is

$$\sigma = 0.173 \times 10^{-8} \text{ Btu/(hr) (sq ft) (deg F abs)}^4 \quad \text{. . . . . (D-3)}$$

The emissivity factor is taken as the product of the respective surface emissivities,

$$F_e = e_1 e_2 \quad \text{. . . . . (D-4)}$$

For simplicity, and also in accord with established data on interior finishes and glass, all emissivities will be taken as 0.92. Thus,

$$F_e = (0.92)^2 = 0.85 \quad \text{. . . . . (D-5)}$$

The room air temperature is to be maintained at 75 F, the interior surfaces will tend to be warmer than the air, and the exposed wall will have a mean temperature appreciably above the room air temperature. Strictly, the initial assumptions for T_{av} should reflect whether or not the outer wall or glass is one of the surfaces involved. For simplicity, however, and entirely within the initial exploratory purposes intended, the assumption will be made that the same T_{av} prevails for all exchanges within the room, namely,

$$T_{av} = (80 + 460) = 540 \text{ deg F abs} \quad \text{. . . . . (D-6)}$$

3. *Expression used in calculations:* Substituting the indicated magnitudes in Equation D-2, the expression for the radiation-exchange resistance becomes

$$R_{(1\rightarrow 2)D} = \frac{1.079}{(F_s A_1)} \quad \text{. . . . . (D-7)}$$

These values are summarized in Table D-1.

4. *Shape factors:* The shape factors needed for Equation D-7 are those for finite rectangular areas oriented in the illustrative room as shown in Fig. 2 of the paper. The assumption will be made that all surfaces radiate diffusely.

Final shape-factor magnitudes are summarized in Table D-1. These were calculated from the equations given by Kollmar,[D-1] and checked with the curves given by Mackey et al.[D-2] An additional useful source was found in illumination flux factors.[D-3] The calculated results meet (very closely) the test that the summation of shape factors for a sequence of surfaces surrounding any emitting surface should be unity.

The so-called reciprocity theorem gives

$$F_{s1}A_1 = F_{s2}A_2 \quad \ldots\ldots\ldots\ldots \quad \text{(D-8)}$$

so that

$$R_{1\rightarrow 2} = R_{2\rightarrow 1} \quad \ldots\ldots\ldots\ldots \quad \text{(D-9)}$$

CONVECTION RESISTANCES

1. *Outdoor surfaces:* Calculations are based upon the assumed magnitude,

$$h_o = 4.0 \text{ Btu/(hr) (sq ft) (F deg)} \quad \ldots\ldots\ldots \quad \text{(D-10)}$$

A convection resistance is, by definition, $1/hA$. The surface areas are

$A_{o,h} = 135$ sq ft
$A_{o,g} = 65$ sq ft

Because the glass surface temperature is introduced directly, the resistance between the outer glass surface and the outdoor air does not enter the calculations. The resistance for the outdoor masonry surface is

$$R = \frac{1}{hA} = \frac{1}{4 \times 135} = 0.00185 \text{ F deg/(Btu/hr)} \quad \ldots\ldots \quad \text{(D-11)}$$

TABLE D-1—RADIATION SHAPE FACTORS AND CIRCUIT RESISTANCES

$$R_{(1\rightarrow 2)D} = \frac{1}{\sigma A_1 F_s F_e A T_{av}^3} = \frac{1.079}{(F_s A_1)}, \text{ F/(Btu/hr)}$$

CIRCUIT QUANTITY*	AREA A_1		F_s	$R_{(1\rightarrow 2)D}$
	Symbol	sq ft		
$R_{e\rightarrow h}$	e	200.	0.1304	0.04139
$R_{e\rightarrow g}$			.0838	.06435
$R_{e\rightarrow f}$			.2993	.01802
$R_{e\rightarrow c}$			.2993	.01802
$R_{e\rightarrow b}$			.0939	.05745
$R_{e\rightarrow d}$			.0939	.05745
$R_{d\rightarrow b}$	d	96	0.0580	0.1938
$R_{d\rightarrow g}$			.0380	.2958
$R_{d\rightarrow h}$			.1575	.07137
$R_{d\rightarrow e}$			.2831	.03970
$R_{d\rightarrow f}$			.2831	.03970
$R_{f\rightarrow h}$	f	300	0.1366	0.02634
$R_{f\rightarrow g}$			.0628	.05727
$R_{f\rightarrow b}$			.0906	.03970
$R_{f\rightarrow e}$			.4200	.00857
$R_{b\rightarrow h}$	b	96	0.1575	0.07137
$R_{b\rightarrow e}$			.2831	.03970
$R_{b\rightarrow g}$			.0380	.2958
$R_{c\rightarrow h}$	c	300	0.1277	0.02817
$R_{c\rightarrow g}$			.0717	.05018

* The shape factor F_s involves direction in that in this tabulation it is the fraction of the radiation emitted by the surface designated by the first subscript that falls on the surface designated by the second subscript. The circuit resistance, $R_{(1\rightarrow 2)D}$, is the same in either direction since it includes the product of F_s and the area appropriate to F_s.

2. *Interior surfaces:* Free convection is considered to be the mechanism for heat transfer between the interior surfaces and the room air. The unit conductance for free convection is known to vary with position and with temperature difference. It has been judged that present purposes will be served adequately, however, by the following approximations:

Horizontal surfaces facing upward,

$$h = 1.1 \text{ Btu/(hr) (sq ft) (F deg)}$$

TABLE D-2—CONVECTION RESISTANCES FOR INTERIOR ROOM SURFACES

$$R = \frac{1}{hA}, \text{ F/(Btu/hr)}$$

CIRCUIT QUANTITY	R
R_{fa}	0.00303
R_{ca}	0.00834
R_{ea}	0.00714
R_{ba}	0.0149
R_{da}	0.0149
R_{ha}	0.0106
R_{ga}	0.0220

TABLE D-3—SUMMARY OF CIRCUIT RESISTANCES AND CAPACITANCES

Branch Resistances x 10^2 in F/(Btu) (hr)
Example: $R_{de} = 0.0575$ deg F/(Btu) (hr)

Branch	b	d	h	e	c	f	m	k	g	s
c	3.97	3.97	2.82	1.80	—	0.857	—	—	5.02	—
d	19.4	—	7.14	5.75	3.97	3.97	—	—	29.58	—
b	—	19.4	7.14	5.75	3.97	3.97	—	—	29.58	—
e	5.75	5.75	4.14	—	1.80	1.80	—	—	6.44	—
f	3.97	3.97	2.63	1.80	0.857	—	—	—	5.73	—
h	7.14	7.14	—	4.14	2.82	2.63	1.03	—	—	—
m	—	—	1.03	—	—	—	—	1.37	—	—
k	—	—	—	—	—	—	1.37	—	—	0.350
a	1.49	1.49	1.06	0.714	0.834	0.303	—	—	2.20	—
To Ground	0.167	0.167	—	0.080	0.068	0.068	—	—	—	—

Capacitances and Reactances
Example: X_{eb} for F = 1/24 hr^{-1} = 0.0289
Capacitance, *C*, Btu/F

BRANCH	b	d	—	e	c	f	m	k
	132.	132.		276.	1020.	1020.	1260.	1150.
Capacitive Reactance x 10^2, X_c*								
f = 1/24 hr^{-1}	2.89	2.89		1.385	0.375	0.375	0.303	0.332
f = 1/12	1.45	1.45		0.693	0.187	0.187	0.152	0.166
f = 1/8	0.965	0.965		0.461	0.125	0.125	0.101	0.111
f = 1/6	0.724	0.724		0.346	0.937	0.937	0.0758	0.0831

* $X_c = \frac{1}{2\pi fC}$ where f = frequency

Horizontal surfaces facing downward,

$$h = 0.40 \text{ Btu/(hr) (sq ft) (F deg)}$$

All vertical surfaces,

$$h = 0.70 \text{ Btu/(hr) (sq ft) (F deg)}$$

The resulting circuit resistances are listed in Table D-2. Should greater accuracy be desired, these magnitudes could be re-adjusted in a second approximation.

CONDUCTION RESISTANCES AND CAPACITANCES

1. *General:* A conduction resistance is calculated from the expression

$$R = \frac{l}{kA}, \text{ F/(Btu/hr)} \quad \text{(D-12)}$$

where

l = length of conductor, ft.
k = thermal conductivity, Btu/(hr) (sq ft) (Fahrenheit deg/ft)
A = cross-sectional area, sq ft.

The thermal capacitance for a particular lumped portion of matter is

$$C = c_p \rho V \text{ Btu/F deg} \quad \text{(D-13)}$$

where

c_p = unit heat capacity at constant pressure, Btu/(lb) (Fahrenheit deg)
ρ = density, lb/cubic ft
V = volume lumped, cubic ft

The length of conductor belonging to a particular resistor in the circuit is established with reference to the suggested rule for lumping, Equation C-2, namely

$$l = 0.44 \sqrt{\frac{a}{f}}, \text{ ft}$$

2. *Floor and Ceiling:* For cinder-aggregate concrete, $a = 0.020$ sq ft/hr. Hence, for the fundamental frequency of 1/24 hr^{-1},

$$l = 0.44 \sqrt{\frac{0.020}{(1/24)}}$$
$$= 0.30 \text{ foot.}$$
$$= 3.6 \text{ in.}$$

Taking a length less than this will improve the accuracy of the analysis.

The floor and ceiling surfaces are not quite interchangeable in the simple problem, but it is a helpful simplification to make the assumption that the mid-planes of the 4-in. slabs comprising the floor and ceiling are planes across which no heat flows at any time. Therefore, the half slab thickness, or 2 in., will be taken as the length of conductor to be lumped into a single capacitance for both floor and ceiling.

With the thermal properties from Table 1 of the paper, the thermal capacitance is,

$$C_f = C_c = 0.21 \times 97 \times \left(300 \times \frac{2}{12} \right) = 1020 \text{ Btu/F deg}$$

This capacitance, connected to ground, is to be attached at the midpoint of the resistance. Thus the effective resistance between c or f in the circuit and the corresponding capacitance is given by a *one-inch thickness* of concrete, or

$$R_c = R_f = \frac{1/12}{0.41 \times 300} = 0.000680 \text{ F/(Btu/hr).}$$

It is proper to note that the lumping adopted becomes more inaccurate for the higher harmonics, because of the increase in frequency. Nothing will be done about this in the present problem.

3. *Interior partitions:* The midsection of the stud space will be taken to be a plane of no heat flow.

The ⅞ in. thickness of gypsum lath and plaster falls easily within the limit for a single lumped capacitance, even with half of the studs added on a stud-volume-per-unit-of-surface-area basis.

The cubic feet of wood in the studs per square foot of partition area, and on one side of the midsection plane, is calculated for 1¾ × 3¾-in. studs on 16-in. centers. This becomes 0.0171 cu ft/sq ft.

With the thermal properties from Table 1 of the paper, the thermal capacitance per square foot of partition surface is the sum of the wood capacitance and the lath-and-plaster capacitance, or

$$C = 0.0171 \times 0.40 \times 30 + 0.23 \times 70 \times \left(\frac{7/8}{12}\right)$$
$$= 0.205 + 1.172 = 1.38 \text{ Btu/(F deg) (sq ft)}$$

Multiplying by the respective partition areas, the circuit quantities become

$C_s = 276$ Btu/F
$C_b = 132$ Btu/F
$C_d = 132$ Btu/F

The thermal capacitance of the air in the stud space is neglected, as are any nonuniformities in partition structure.

Because of the studs, the circuit resistance between the partition surface and the point of connecting the capacitance corresponds to slightly more than one half of the lath-and-plaster thickness. Accurate calculation of this is not justified for present purposes, and so the very rough guess will be made that 60 percent of the lath-and-plaster thickness will give a representative point.

Thus, per square foot of partition area,

$$R/\text{sq ft} = 0.6 \times \left(\frac{7/8}{12}\right) \times \frac{1}{0.275} = 0.16 \text{ F deg/(Btu/(hr) (sq ft)}$$

The circuit resistances for the appropriate areas then are as follows:

$R_s = 0.000800$ F deg/(Btu/hr)
$R_b = 0.00167$ F deg/(Btu/hr)
$R_d = 0.00167$ F deg/(Btu/hr)

4. *Exterior wall: (a) Lumping:* The suggested length of brick to be lumped is determined for $a = 0.023$ sq ft/hr, and $f = 1/24$ hr^{-1}, as

$$l = 0.44 \sqrt{\frac{0.023}{(1/24)}}$$
$$= 0.327 \text{ ft.}$$
$$= 3.92 \text{ in.}$$

For the concrete blocks, this length is calculated to be 4.7 in., but this result must be *strongly qualified* by the fact that the conductivity and density given in Table 1 of the paper both have been averaged over the core holes which make up a large proportion of the volume of a concrete block. Hence, any temperature calculated for a point within the hypothesized continuous medium replacing the blocks will have an uncertain physical significance.

Another type of approximation for the blocks would be to replace them with a thin slab of a continuous medium having the same thermal conductivity and density as the

concrete itself. Such a slab would be roughly of the thickness suggested by the lumping rule. Moreover, the lumping rule was derived for a continuous medium.

(A further practical consideration in the lumping of the outer wall is that each junction point added to the circuit introduces another equation and another unknown. While this may be a very minor matter for machine calculation, it is significant for algebraic or numerical work.)

The final decision in setting up this illustrative problem was to divide the outer wall into two capacitance lumps, as follows: first, or outer unit, comprising the brick and mortar, and second, or inner unit, comprising the concrete blocks and inside finish.

A slight compromise in final accuracy is accepted by employing two lumps instead of three, but the initial exploratory intent is served equally well.

(b) The outer lumped unit: The capacitance is the sum of the $c_p\rho V$ products for the brick and the mortar, or C_k in the circuit shown in Fig. 2 of the paper is

$$C_k = 0.20 \times 112 \times \frac{4}{12} \times 135 + 0.22 \times 115 \times \frac{1/2}{12} \times 135$$
$$= 1008 + 142 = 1150 \text{ Btu/F}$$

The thermal resistance for this portion of the path comprises the outer surface conductance, the brick, and the mortar in series, or

$$R = \frac{1}{hA} + \sum\left(\frac{l}{kA}\right) = \frac{1}{4 \times 135} + \frac{4}{12 \times 0.51 \times 135} + \frac{1/2}{12 \times 1.0 \times 135}$$
$$= 0.00185 + 0.00484 + 0.00031 = 0.00700 \text{ F deg/(Btu/hr)}$$

Placing C_k at the midpoint of this resistance, the circuit quantity R_{ak} is

$$R_{ak} = 0.00350 \text{ F/(Btu/hr)}$$

The physical location of point k is obtained by proportioning the part of this resistance caused by the brick to the total brick resistance. Thus, the distance in from the outer surface becomes

$$\frac{0.00350 - 0.00185}{0.00484} \times 4 = 1.36 \text{ in.}$$

(Afterthought, following solution of the circuit, indicated that it would have been preferable to have placed k at the physical midpoint, but the solution was not modified to make this change on the basis that the exploratory intent would not have been aided sufficiently to compensate for the extra calculation required.)

(c) The inner lumped unit: The capacitance C_m in the circuit is the sum of the $c_p\rho V$ products for the blocks and the interior finish. The furring strips for the lath and plaster will be assumed as nominal 2×2's so that the interior finish capacitance is 1.38 Btu/(F deg) (sq ft) as calculated in Section 3 of the part Conduction Resistances and Capacitances in this Appendix. Hence,

$$C_m = 0.21 \times 57 \times 8/12 \times 135 + 1.38 \times 135$$
$$= 1260 \text{ Btu/F}$$

The total resistance of this unit is the sum of the contributions from the blocks, the furring space, and the lath and plaster. When the conductance of the furring space is taken as 1.2 Btu/(hr) (sq ft) (F deg), which includes an allowance for the presence of the furring strips, the resistance becomes

$$R = \left(\frac{l}{kA}\right)_{\text{block}} + \left(\frac{1}{1.2A}\right)_{\text{space}} + \left(\frac{l}{kA}\right)_{\text{lath + plaster}}$$
$$= \frac{8}{12 \times 0.30 \times 135} + \frac{1}{1.2 \times 135} + \frac{7/8}{12 \times 0.275 \times 135}$$
$$= 0.01235 + 0.00617 + 0.00196 = 0.02048 \text{ F deg/(Btu/hr)}$$

The circuit resistance R_{km} is composed of half of the resistance for the outer lump plus half of the resistance for the inner lump, or

$$R_{km} = 0.00350 + 0.01024 = 0.0137 \text{ F deg/(Btu/hr)}$$

The circuit resistance R_{mh} is half of the resistance of the inner lump, or

$$R_{mh} = 0.0102 \text{ F deg/(Btu/hr)}$$

Attempting to locate point m in the physical structure is dubious because of the core holes in the cement blocks.

Circuit resistances and capacitances are summarized in Table D-3.

CAPACITIVE REACTANCES

The capacitive reactances were calculated from the defining relation,

$$X_c = \frac{1}{2\pi f C}$$

as noted in Table D-3 of the paper.

DISCUSSION

The solution given in the paper describes the behavior of the network system as defined by Fig. 3 and the assigned magnitudes of resistance, capacitance, and input potentials. The important unanswered question is how closely the network, as defined, may represent the structure which served as its prototype. Present experience is inadequate to discuss this point. Compromises and approximations were freely made in establishing the circuit. Refinements may be made as desired. For instance, T_{sv} in the radiation resistance could be re-adjusted, the outer wall could be lumped into three parts instead of two, and the estimated free-convection conductances could be readjusted on the basis of the first-approximation temperatures obtained.

Certainly, the experience gained in this initial exploration has been an invaluable first step towards whatever future studies may be justified.

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APPENDIX E

Circuit Input Potentials, Equations to be Solved, and Solutions Obtained

The input potentials are given in the paper as follows: Fig. 4, Sol-Air Temperature; Fig. 5, Indoor Surface Temperature for Single Glass, and Indoor Surface Temperature for Glass Block.

The Fourier-series expressions for these curves are summarized in Table E-1. The fourth harmonic was judged to be the limit of significance.

Current-balance equations are written in accord with Appendix B for each junction point. Taking $T_a = 0$ as a datum (ground), gives the set of simultaneous equations in Table E-2. The numerical coefficients come from Appendix D. The complex quantities (those involving the vector operator j) are determined in accord with Table A-1 in Appendix A.

The complete set of equations was found to be overly cumbersome for solution by conventional algebraic methods. Therefore, the *real* and *imaginary* parts of the vector terms were separated, yielding two sets of equations instead of one and the solution was found through use of a computing machine designed especially to handle simultaneous linear equations.

These results gave the temperature at each junction point in the system. The initial vector-form results have been expressed in Fourier series terms as given in Table E-3. These quantities, when plotted, give Figs. 9 and 10 in the paper.

The instantaneous system load is the current q_s in Section B of Table E-2. Results are tabulated in Table E-4 and plotted in Fig. 6 of the paper.

The thermal current in any branch of the circuit is obtained from the relation

$$q = Y(\Delta T), \text{ Btu/hr} \quad \ldots\ldots\ldots\ldots \text{(E-1)}$$

where

Y = admittance for the branch involved, Btu/(hr) (F)
ΔT = potential difference across the branch involved, (F)

The currents of prime interest are summarized in Table E-5 and plotted in Figs. 7 and 8 of the paper.

Numerical example: The manipulation of the equation for point k will be given as a numerical example. From Table E-2,

$$Y_{sk}\,T_s + Y_{km}\,T_m - (Y_{sk} + Y_{km} + Y_{Ck})\,T_k = 0 \quad \ldots\ldots \text{(E-2)}$$

$$\text{Let } K = -Y_{sk} - Y_{Ck} - Y_{km} \quad \ldots\ldots\ldots\ldots \text{(E-3)}$$

the admittances are obtained from Appendix D, Table D-3.

TABLE E-1—CIRCUIT INPUT POTENTIALS

Fourier-series expressions for the curves of Figs. 4 and 5 of the paper have the following form, with the datum being the indoor air temperature of 75 F.

$$T = (a_0 - 75) + a_1 \cos 15\Theta + b_1 \sin 15\Theta + a_2 \cos 30\Theta + b_2 \sin 30\Theta + a_3 \cos 45\Theta + b_3 \sin 45\Theta + a_4 \cos 60\Theta + b_4 \sin 60\Theta + \text{- - -}$$

where

Θ = elapsed time in hours after midnight, sun time 15Θ, 30Θ, 45Θ, 60Θ are the equivalent angular degrees for the fundamental, second, third, and fourth harmonics, respectively.

Note that the corresponding vector notation for each harmonic $= a - jb$.

	SOL-AIR TEMPERATURE F	SINGLE GLASS INDOOR SURFACE TEMPERATURE F	GLASS BLOCK INDOOR SURFACE TEMPERATURE F
a_1	−9.47	−4.49	−1.30
b_1	−8.23	−5.92	−4.72
a_2	−0.959	0.538	−1.11
b_2	1.72	1.29	0.790
a_3	0.904	−0.172	−0.0583
b_3	0.0878	−0.0142	−0.0167
a_4	1.82	0.0833	0.538
b_4	−0.0866	−0.0433	0.238
a_0	86.5	80.8	79.0

TABLE E-2—CIRCUIT EQUATIONS TO BE SOLVED

A. GENERAL FORM (*Top*) B. NUMERICAL FORM (*Center*) C. VALUES OF THE COMPLEX QUANTITIES C, D, E, F, H, M, K. (*Bottom*)

CIRCUIT POINT	T_b	T_d	T_h	T_e	T_c	T_f	T_m	T_k	T_g	T_a
c	$Y_{cb}\,T_b$	$+Y_{cd}\,T_d$	$+Y_{ch}\,T_h$	$+Y_{ce}\,T_e$	$-(\Sigma Y_{nc})T_c$	$+Y_{cf}\,T_f$	$+0$	$+0$ =	$-Y_{cg}\,T_g$	$+0$
d	$Y_{db}\,T_b$	$-(\Sigma Y_{nd})T_d$	$+Y_{dh}\,T_h$	$+Y_{de}\,T_e$	$+Y_{dc}\,T_c$	$+Y_{df}\,T_f$	$+0$	$+0$ =	$-Y_{dg}\,T_g$	$+0$
b	$-(\Sigma Y_{nb})T_b$	$+Y_{bd}\,T_d$	$+Y_{bh}\,T_h$	$+Y_{be}\,T_e$	$+Y_{bc}\,T_c$	$+Y_{bf}\,T_f$	$+0$	$+0$ =	$-Y_{bg}\,T_g$	$+0$
e	$Y_{eb}\,T_b$	$+Y_{ed}\,T_d$	$+Y_{eh}\,T_h$	$-(\Sigma Y_{ne})T_e$	$+Y_{ec}\,T_c$	$+Y_{ef}\,T_f$	$+0$	$+0$ =	$-Y_{eg}\,T_g$	$+0$
f	$Y_{fb}\,T_b$	$+Y_{fd}\,T_d$	$+Y_{fh}\,T_h$	$+Y_{fe}\,T_e$	$+Y_{fc}\,T_c$	$-(\Sigma Y_{nf})T_f$	$+0$	$+0$ =	$-Y_{fg}\,T_g$	$+0$
h	$Y_{hb}\,T_b$	$+Y_{hd}\,T_d$	$-(\Sigma Y_{nh})T_h$	$+Y_{he}\,T_e$	$+Y_{hc}\,T_c$	$+Y_{hf}\,T_f$	$+Y_{hm}\,T_m$	$+0$ =	0	$+0$
m	0	$+0$	$+Y_{mh}\,T_h$	$+0$	$+0$	$+0$	$-(\Sigma Y_{nm})T_m$	$+Y_{mk}\,T_k$ =	0	$+0$
k	0	$+0$	$+0$	$+0$	$+0$	$+0$	$+Y_{km}\,T_m$	$-(\Sigma Y_{nk})T_k$ =	0	$-Y_{ks}\,T_s$
a	$Y_{ab}\,T_b$	$+Y_{ad}\,T_d$	$+Y_{ah}\,T_h$	$+Y_{ae}\,T_e$	$+Y_{ac}\,T_c$	$+Y_{af}\,T_f$	$+0$	$+0$ =	$-Y_{ag}\,T_g$	$+0+q_aL.$

Notes: (1) Choice of T_a as datum makes $T_a = 0$. (2) $T_b = T_d$ from the symmetry of the physical system. In substituting numerical values for the coefficients (see Part B of this table), the two equations have been added so that only one equation, that for circuit point *d*, is given. (3) Transmitted radiation and infiltration are not included in these equations.

CIRCUIT POINT	T_d	T_h	T_e	T_c	T_f	T_m	T_k	T_g	T_a
c	$50.4T_d$	$+35.5T_h$	$+55.5T_e$	$+C\,T_c$	$+117.6T_f$	$+0$	$+0$ =	$-19.9T_g$	$+0$
d	$D\,T_d$	$+14.0T_h$	$+17.4T_e$	$+25.2T_c$	$+25.2T_f$	$+0$	$+0$ =	$-\ 3.38T_g$	$+0$
e	$34.8T_d$	$+21.2T_h$	$+E\,T_e$	$+55.5T_c$	$+55.5T_f$	$+0$	$+0$ =	$-15.5T_g$	$+0$
f	$50.4T_d$	$+38.0T_h$	$+55.5T_e$	$+117.6T_c$	$+F\,T_f$	$+0$	$+0$ =	$-17.6T_g$	$+0$
h	$28.0T_d$	$+H\,T_h$	$+24.2T_e$	$+35.5T_c$	$+38.0T_f$	$+97.1T_m$	$+0$ =	0	$+0$
m	0	$+97.1T_h$	$+0$	$+0$	$+0$	$+MT_m$	$+73.4T_k$ =	0	$+0$
k	0	$+0$	$+0$	$+0$	$+0$	$+73.4T_m$	$+K\,T_k$ =	0	$-294.T_s$
	$*134.2T_d$	$*+94.4T_h$	$*+140.0T_e$	$*+119.9T_c$	$*+330.0T_f$	$*+0$	$*+0$ =	$*-45.5T_g$	$*0+q_aL.$

HARMONIC	C	D	E	F	H	M	K
a_0 term or *dc* component	-399	-152	-326	-609	-317	-171	-368
1	$-446-j258$	$-155-j34.6$	$-330-j72.0$	$-656-j258$	$-317+j0$	$-171-j330$	$-368-j301$
2	$-570-j472$	$-161-j69.0$	$-342-j143$	$-780-j472$	$-317+j0$	$-171-j660$	$-368-j602$
3	$-735-j618$	$-170-j104$	$-362-j210$	$-946-j618$	$-317+j0$	$-171-j990$	$-368-j903$
4	$-906-j699$	$-183-j138$	$-389-j274$	$-1117-j699$	$-317+j0$	$-171-j1320$	$-368-j1204$

* Point *a* equation for use in load calculation.

$$Y_{sk} = \frac{1}{R_{sk}} = \frac{1}{0.350 \times 10^{-2}} = 294.$$

$$Y_{km} = \frac{1}{R_{km}} = \frac{1}{1.37 \times 10^{-2}} = 73.4$$

For the first harmonic only (see Table A-1, Appendix A),

$$Yc_k = \frac{1}{-jXc_k} = \frac{j}{Xc_k} = \frac{j}{= 0.332 \times 10^{-2}} = j\,301$$

therefore.

$$K = -(294 + 73.4 + j\,301) = -(368 + j\,301)$$

For the first harmonic only, in vector notation, Equation E-2 becomes

$$294\,T_s + 73.4\,T_m - (368 + j\,301)\,T_k = 0 \quad \ldots\ldots\ldots \quad \text{(E-4)}$$

the temperatures for the first harmonic are:

$$T_s = (a_o - 75)_s + (a_1 - j\,b_1)_s$$

$$T_k = (a_o - 75)_k + (a_1 - j\,b_1)_k$$

$$T_m = (a_o - 75)_m + (a_1 - j\,b_1)_m$$

The coefficients a and b for T_s are given in Table E-1.

Drop the subscript "one" for convenience and drop the a_o terms for subsequent consideration. Substitutions into Equation E-4 yield the following relations for the

TABLE E-3—SOLUTIONS OF THE CIRCUIT EQUATIONS OF TABLE E-2

Solutions are constants for the Fourier series expressions of temperature, T, where:

$$T = a_o + a_1 \cos 15\Theta + b_1 \sin 15\Theta + a_2 \cos 30\Theta + b_2 \sin 30\Theta + a_3 \cos 45\Theta + b_3 \sin 45\Theta + a_4 \cos 60\Theta + b_4 \sin 60\Theta$$

where:

Θ = elapsed time in hours after midnight.

CASE I	a_o	a_1	b_1	a_2	b_2	a_3	b_3	a_4	b_4
$T_b = T_d$	75.6203	+0.031	−0.329	−0.0142	+0.0397	−0.0032	−0.0036	+0.0013	+0.0006
T_c	75.8086	+0.087	−0.424	−0.0229	+0.0427	−0.0026	−0.0037	+0.0016	+0.0005
T_e	75.7212	−0.051	−0.476	−0.0129	+0.0689	−0.0059	−0.0053	+0.0029	+0.0007
T_f	75.5633	+0.040	−0.307	−0.0144	+0.0339	−0.0025	−0.0028	+0.0013	+0.0003
T_h	76.9623	+0.437	−0.402	−0.0139	−0.0088	−0.0068	+0.0010	−0.0058	+0.0032
T_k	85.2907	−1.064	−7.607	−0.8138	+0.0262	+0.0757	+0.2583	+0.1393	+0.4011
T_m	80.5315	+1.383	−0.825	−0.0240	−0.0863	−0.0179	+0.0080	−0.0212	+0.0100

CASE II	a_o	a_1	b_1	a_2	b_2	a_3	b_3	a_4	b_4
$T_b = T_d$	75.5122	+0.123	−0.212	−0.0420	−0.0083	−0.0011	−0.0016	+0.0028	+0.0118
T_c	75.6464	+0.180	−0.246	−0.0455	−0.0150	−0.0008	−0.0016	+0.0038	+0.0115
T_e	75.5730	+0.106	−0.318	−0.0674	−0.0051	−0.0019	−0.0023	+0.0081	+0.0197
T_f	75.4535	+0.117	−0.188	−0.0356	−0.0084	−0.0008	−0.0012	+0.0036	+0.0090
T_h	76.8978	+0.474	−0.341	−0.0248	−0.0309	−0.0059	+0.0008	−0.0049	+0.0080
T_k	85.2829	−1.068	−7.610	−0.8131	+0.0265	+0.0758	+0.2563	+0.1392	+0.4013
T_m	80.4915	+1.373	−0.810	−0.0213	−0.0885	−0.0179	+0.0081	−0.0216	+0.0101

unknowns T_k and T_m.

$$294\,(-9.47 + j\,8.23) + 73.4\,a_m - j\,73.4\,b_m - 368\,a_k + j\,368\,b_k - j\,301\,a_k - 301\,b_k = 0 \quad \text{(E-5)}$$

By grouping the j terms and the terms which do not contain j the following two equations result

$$73.4\,a_m - 368\,a_k - 301\,b_k = +2784$$
$$-73.4\,b_m - 301\,a_k + 368\,b_k = -2420$$

Other equations are developed similarly. The final set of simultaneous equations then is solved for the unknown a and b coefficients at each circuit junction point.

The a_o terms give the steady-state solution. In the steady-state, the equation for point k is

$$294\,(86.5 - 75) + 73.4\,(a_o - 75)_m - 368\,(a_o - 75)_k = 0 \quad \text{(E-6)}$$

$a_o = 86.5$ refers to T_s. Recall that the datum or zero potential was taken as 75 F; hence, this can be dropped from the equation, giving

$$294\,(86.5) + 73.4\,(a_o)_m - 368\,(a_o)_k = 0 \quad \text{(E-7)}$$

To check the dropping of the 75 from equation (E-6),

$$-294 - 73.4 + 368 = 0.6 \cong 0.$$

The remainder of 0.6 simply is an arithmetical inaccuracy which is not important.

TABLE E-4—INSTANTANEOUS SYSTEM LOAD

Heat transferred to the air stream a by convection from surfaces b, c, d, e, f, g and h

SUN TIME	CASE I SINGLE WINDOW GLASS BTU/HR	CASE II GLASS BLOCK BTU/HR
12 Midnight	763	747
1 a.m.	673	667
2	596	576
3	515	506
4	427	460
5	382	410
6	361	460
7	347	421
8	398	353
9	513	342
10	666	388
11	883	479
12 noon	1101	608
1 p.m.	1276	751
2	1406	886
3	1499	985
4	1539	1079
5	1540	1174
6	1501	1280
7	1348	1256
8	1264	1154
9	1154	1023
10	1018	909
11	892	838

Re-examination of the two-capacitance lumping of the outer wall: In examining the results obtained, it was decided to briefly investigate the consequences of employing *three* capacitance lumps instead of two in the outer-wall circuit branch, s to h in Fig. 3 of the paper. Three capacitances would be an improvement in accuracy but the magnitude was uncertain.

The wall was divided into three equal capacitance lumps and, to greatly simplify the work, the problem was re-solved under the assumption that all indoor surface temperatures except T_h and T_g were *constant* at their previously determined average magnitudes. This gave only four unknown temperatures and permitted the use of determinants for calculations. Findings for the indoor surface temperature of the outer wall are compared with values obtained in the same manner for a two-lump circuit and are as follows:

	TWO LUMPS		THREE LUMPS	
	a_h	b_h	a_h	b_h
1st harmonic	0.684	−0.219	0.693	−0.187
2nd harmonic	0.264	−0.015	0.285	−0.022

Only two harmonics were investigated, and the results indicate that a finer subdivision for the capacitance lumps *does* improve the accuracy; however, the change for present purposes is not of sufficient magnitude to either cause concern or to greatly shift the system load. Therefore, the matter was not carried further. The recommendation is that, for more accurate work, the lumping rule in Appendix C should be followed closely.

APPENDIX F

Periodic Heat Gain by the Guide Method

Wall Section: The treatment of the cycle of instantaneous heat-gain rate given here follows Chapter 12 on Cooling Load of the Heating, Ventilating and Air Conditioning GUIDE, 1952 Edition. Equation 5 of this chapter gives, for wall sections,

$$\frac{q}{A} = U\,(t_m - t_i) + \lambda\, U\,(t_e^* - t_m), \text{ Btu/(hr) (sq ft)} \quad . \quad . \quad . \quad . \quad . \quad \text{(F-1)}$$

where

U = over all conductance of the wall, Btu/(hour) (square foot) (Fahrenheit)
t_m = 24 hour average sol-air temperature, Fahrenheit
t_i = constant indoor air temperature, Fahrenheit
λ = decrement factor, dimensionless
t_e^* = sol-air temperature at a time in advance of the time being considered by the amount of the time lag involved, Fahrenheit

From the given data,

t_m = 86.5 F
t_i = 75. F

The over-all conductance is:

$$\frac{1}{U} = \left(\frac{1}{h}\right)_{\text{outer surface}} + \left(\frac{l}{k}\right)_{\text{brick}} + \left(\frac{l}{k}\right)_{\text{mortar}} + \left(\frac{l}{k}\right)_{\text{blocks}} + \left(\frac{1}{C}\right)_{\text{air space}} + \left(\frac{l}{k}\right)_{\text{lath and plaster}} + \left(\frac{1}{h}\right)_{\text{inner surface}}$$

where

h = unit surface conductance, Btu/(hour) (square foot) (Fahrenheit degree)
l = heat flow path length, foot
k = thermal conductivity, Btu/(hour) (square foot) (Fahrenheit/foot)
C = air space conductance, Btu/(hour) (square foot) (Fahrenheit)

Table E-5—Instantaneous Thermal Currents of Interest

Case I—Single Window Glass *(All values in Btu/hr)*

Sun Time	Heat Balance at Point *m*			Heat Balance at Point *d*		
	Conduction *k* to *m*	Rate of Heat Storage at *m*	Conduction *m* to *h*	Gain by *d* in Net Radiant Exchanges	Convection *d* to Room Air, *a*	Rate of Heat Storage at *d*
12 M	127	−308	435	33	42	−9
1 a.m.	49	−371	420	29	38	−9
2	− 41	−443	402	26	34	−8
3	−124	−504	380	23	30	−7
4	−169	−524	355	19	27	−8
5	−156	−485	329	17	24	−7
6	− 92	−397	305	17	21	−4
7	− 10	−296	286	16	20	−4
8	+ 60	−212	272	19	20	−1
9	126	−135	261	26	23	+3
10	217	− 42	253	33	27	6
11	331	+ 81	250	44	33	11
12 noon	479	226	253	54	40	14
1 p.m.	619	355	264	61	47	14
2	730	449	281	65	53	12
3	813	510	303	68	59	9
4	877	548	329	69	63	6
5	922	564	358	68	65	3
6	923	536	387	67	65	2
7	857	442	415	57	63	−6
8	707	270	437	55	60	−5
9	522	73	449	50	56	−6
10	348	−105	453	45	52	−7
11	218	−229	447	39	47	−8

Case II—Glass Block *(All values in Btu/hr)*

12 M	133	−300	433	33	40	−7
1 a.m.	54	−363	417	29	37	−8
2	− 39	−436	397	24	33	−9
3	−128	−503	375	22	30	−8
4	−176	−525	349	21	27	−6
5	−159	−482	323	18	25	−7
6	− 89	−389	300	22	23	−1
7	− 6	−289	283	19	22	−3
8	+ 64	−206	270	16	21	−5
9	124	−139	263	15	19	−4
10	211	− 46	257	19	19	0
11	336	+ 79	257	25	21	+4
12 noon	488	227	261	31	25	6
1 p.m.	632	358	274	38	29	9
2	740	447	293	44	34	10
3	816	501	315	48	39	9
4	879	538	341	52	43	9
5	924	557	367	56	47	9
6	931	536	395	61	51	10
7	864	444	420	58	54	4
8	714	273	441	50	53	−3
9	520	69	451	44	51	−7
10	344	−109	453	39	47	−8
11	219	−227	446	37	43	−6

TABLE E-5 *(Cont'd)*

CASE I—SINGLE WINDOW GLASS *(All values in Btu/hr)*

Sun Time	Heat Balance at Point *e*			Heat Balance at Point *f*		
	Gain by *e* in Net Radiant Exchanges	Convection *e* to Room Air, *a*	Rate of Heat Storage at *e*	Gain by *f* in Net Radiant Exchanges	Convection *f* to Room Air, *a*	Rate of Heat Storage at *f*
12 M	61	92	−31	121	195	−74
1 a.m.	50	82	−32	107	172	−65
2	47	70	−23	96	152	−56
3	40	60	−20	81	134	−53
4	29	52	−23	63	117	−54
5	28	44	−16	57	102	−45
6	33	39	− 6	59	92	−33
7	33	37	− 4	58	86	−28
8	43	39	+ 4	77	86	− 9
9	70	48	22	111	96	+15
10	93	64	29	150	116	34
11	127	85	42	209	142	67
12 noon	157	108	49	261	175	86
1 p.m.	176	130	46	300	206	94
2	185	148	37	324	235	89
3	189	161	28	338	260	78
4	186	170	16	337	279	58
5	180	172	8	328	289	39
6	170	169	1	312	292	20
7	132	161	−29	258	287	−29
8	125	148	−23	238	275	−37
9	112	135	−23	205	264	−59
10	94	120	−26	179	238	−59
11	76	107	−31	151	216	−65

CASE II—GLASS BLOCK *(All values in Btu/hr)*

12 M	67	86	−19	130	177	−47
1 a.m.	60	76	−16	114	163	−49
2	50	66	−16	92	146	−54
3	46	55	− 9	82	129	−47
4	46	49	− 3	76	116	−40
5	34	47	−13	64	107	−43
6	52	47	+ 5	88	101	−13
7	46	47	− 1	79	95	−16
8	24	44	−20	58	88	−30
9	27	41	−14	59	82	−23
10	41	41	0	78	81	− 3
11	59	49	+10	105	89	+16
12 noon	88	62	26	138	106	32
1 p.m.	126	76	50	175	125	50
2	120	89	31	207	146	61
3	130	100	30	229	165	64
4	138	113	25	247	185	62
5	144	126	18	265	205	60
6	161	137	24	291	223	68
7	143	139	4	272	232	40
8	117	134	−17	231	232	− 1
9	95	122	−27	195	221	−26
10	81	108	−27	165	206	−41
11	79	96	−17	152	191	−39

Substituting,

$$\frac{1}{U} = \frac{1}{4} + \frac{4}{12 \times 0.51} + \frac{1}{2 \times 12 \times 1.0} + \frac{8}{12 \times 0.40} + \frac{1}{1.2} + \frac{7}{8 \times 12 \times 0.275} + \frac{1}{1.65} = 4.316$$

$$\therefore U = 0.232 \text{ Btu/(hr) (sq ft) (F deg)}$$

Time Lag: First determine the fundamental-frequency time lag for each wall layer, employing Fig. 2 in the 1944 paper of Mackey and Wright.[F-1]

LAYER	h/l	$k\rho c_p$	FUNDAMENTAL LAG, HR
Brick	1.53	11.4	2.25
Mortar	24.0	25.3	0.19
Blocks	0.60	4.79	3.60
Lath and Plaster	3.77	4.43	0.31
			Total...6.35

TABLE F-1—INSTANTANEOUS HEAT FLOW RATES FOR ILLUSTRATIVE PROBLEM Calculated by THE GUIDE Method

SUN TIME	HEAT FLOW RATE, BTU/HR				
	Outer Wall Total	Single Glass Total	Wall + Glass Total	Glass Block Total	Wall + Block Total
12 Midnight	520	117	637	156	676
1 a.m.	531	52	583	91	622
2	377	33	410	33	410
3	345	0	345	13	358
4	321	−46	275	13	344
5	301	−65	236	−65	236
6	281	0	281	130	411
7	257	0	257	195	452
8	246	130	376	130	376
9	246	325	571	130	376
10	235	520	755	260	495
11	224	845	1069	390	614
12 noon	235	1105	1340	520	755
1 p.m.	345	1300	1645	650	995
2	321	1365	1686	715	1036
3	301	1430	1731	845	1146
4	345	1365	1710	845	1190
5	377	1300	1677	845	1222
6	421	1170	1591	975	1396
7	465	845	1310	845	1310
8	487	715	1202	585	1072
9	498	520	1018	390	888
10	509	384	893	320	829
11	487	220	707	234	721
12 Midnight	520	117	637	156	676

F-1 Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293).

The total of the individual lags is less than the composite-wall lag. Following THE GUIDE suggestion the composite-wall lag will be taken as *7 hours.*

Decrement Factor: Using Fig. 2, page 277, of THE GUIDE 1952, the decrement factor is

$$\lambda = 0.35$$

Instantaneous Heat Flow Rate Through Outer Wall: The outer-wall area is 135 sq ft. Then, from Equation F-1,

$$q = 135 \times 0.232\ (86.5 - 75.0) + 135 \times 0.35 \times 0.232\ (t_e^* - 86.5)$$

$$= 361 + 11.0\ (t_e^* - 86.5) \text{ Btu/hour}$$

The sol-air temperature cycle is given as Table 9, Chapter 12, GUIDE.

Instantaneous Heat Flow Rates Through Glass: The cycles of glass heat flow rates are given directly in THE GUIDE. With a glass area of 65 sq ft, the tabulated data are obtained as follows:

Single glass, Table 17, Chapter 12

Glass Block, Type I, Table 24, Chapter 12

Summary: Calculated heat flow rates are summarized in Table F-1.

APPENDIX G

Treatment of Directly Transmitted Radiation and Radiant Energy Absorption by Room Air

DIRECTLY TRANSMITTED RADIATION

This topic has been reserved for detailed treatment in a future paper. The intended general approach is noted here for completeness.

Net directly transmitted radiant energy may be divided into two components, according to whether the primary source is (1) the sun, or (2) the sky and the environment.

A. *Radiation from the Sun:* The geometry of the room and window, the transmitting characteristics of the glass, and the position of the sun throughout the design day will serve to establish the following: (1) the instantaneous rate of transmitted radiation, Btu/hr, entering the room, and (2) the proportion of the total energy rate in this *beam* which falls directly upon the various room surfaces.

In connection with the second item, it is proposed to adopt the simplifying assumption that the beamed radiation striking any room surface is absorbed with that surface temperature remaining perfectly uniform over its area but variable with time. This, then, will give the thermal-current inputs for the solar component at the respective circuit junctions in Fig. 3 of the paper.

B. *Radiation from Sky and Environment:* It is proposed to treat this radiant flux as being uniformly distributed over the hemispherical half-space defined by the plane of the outer building surface. The geometry of the glass installation and of the room then will establish the proportion of the net transmitted radiation from this source which falls upon the individual room surfaces. As before, the room surfaces will be treated as remaining respectively at uniform time-variable temperatures.

This approach will yield the second component of the transmitted radiation entering the circuit junctions in Fig. 3 of the paper.

C. *Input Currents for the Circuit Equations:* The respective thermal-current inputs from these sources are to be expressed as Fourier series. The coefficients of the series terms, then, will enter the vector equations as constant factors.

ABSORPTION OF RADIANT ENERGY BY ROOM AIR

Referring to the circuit diagram of Fig. 3 of the paper, consider the branches carrying radiant-energy thermal currents. Air absorption may be represented as a circuit leak to ground, wherein the leak resistance is sized in accord with the fraction of the net radiant exchange (calculated with absorption neglected) which is absorbed.

A circuit branch representing the absorption effect will go to ground in parallel with a convection-current branch. Hence, the effect of absorption may be dealt with, in the type of circuit where the air is held at ground potential, by making a slight change in the magnitudes of the proper convection-path conductances.

The radiation thermal current between the room air and a room surface may be calculated according to the method presented by McAdams[G-1]. From this, the circuit conductance is established. The cyclic variation of room surface temperature produces a corresponding conductance variation, which needs to be considered with care. It may be expected, however, and especially in view of the approximate magnitudes of the convective conductances, that a reasonable approximation would be attainable.

The entire practical question of air absorption needs to be investigated in a planned sequence of calculations. To really approach this properly, moisture transfer should also be introduced as part of the thermal circuit.

DISCUSSION

W. R. BRISKEN*, Bloomfield, N. J. (WRITTEN): The authors can be commended for giving a very instructive introduction of circuit techniques for application to cooling load calculations. The example given shows clearly how effects of even relatively small magnitude can be studied, and how information on individual details of interest can be obtained. Undoubtedly, further studies utilizing circuit techniques will reveal interesting facts with respect to equipment characteristics and control functions.

Our department has been using circuit techniques in cooling load calculations for a considerable period of time. One of the purposes was to obtain better data upon which to base the required capacity of cooling equipment for residential applications. While load factors in earlier calculation forms took into account some transient effects, some results of newer studies employing circuit techniques have been entered recently in our 1954 heat loss estimate form.

It was recognized in the early stages of our studies that the greatest danger in application of circuit techniques is the temptation to simulate actual conditions too accurately in the thermal circuit. The circuit becomes too complicated, load calculations become practically impossible, and analogue computations become difficult to control; aside from the fact that elaborate computations require more expensive computers. Therefore, the circuit for a typical one-story house with basement and attic was limited to five branches, one each for walls (all walls combined in one), ceiling plus attic and roof, floor plus basement, windows plus doors and infiltration, and interior partitions plus furniture. It has been proved mathematically that by disregarding radiation from interior surfaces of walls and considering uniform temperature distribution in the air throughout the main story of the house, all exterior walls can be combined into one branch. The paper also indicates that the omission of radiation effects of interior surfaces of walls is justified. The storage of radiated energy (except from direct solar radiation) in partitions and furniture is of very low magnitude especially in comparison with the storage effect existing under varying air temperatures encountered under thermostatic control of the cooling equipment. This storage effect has been found to be of second power magnitude as compared to storage of radiated heat.

G-1 *Heat Transmission*, by W. H. McAdams, Chapter 3 (McGraw-Hill Book Company, New York, Second Edition, 1942).

* Manager, Advance Engineering, General Electric Co.

In our studies all circuit analyses made have been conducted on analogue computers. Signal generators were used to establish the potential corresponding to outdoor temperature. Solar and diffuse radiation were applied where they would be effective, as for example at the outside surface of the walls underneath the film. In many cases studied, instantaneous heat flows through the individual branches of the circuit and the operating time of equipment were recorded graphically, in addition to indoor temperature and total load.

One particular study may be of interest at this point. The basic five-branch circuit we have described was set up for the Research Residence No. 2 of the University of Illinois, and was operated on an analogue computer for essentially the same outdoor weather conditions as those prevailing on the test day of July 22, 1952‡. The computed maximum cooling load was approximately 7 percent higher than the actual cooling load determined by test at the University of Illinois. This result compares favorably with approximately 41 percent excess capacity which would have resulted from maximum cooling load calculations in accordance with the method shown in Chapter 13 of THE GUIDE, 1953 for prevailing conditions of approximately 22 percent for design conditions.

A great advantage of circuit studies is that a mechanical engineer can very readily develop a great ability to express thermal problems by circuits and to draw such circuits with a high degree of accuracy. He does not need to limit his imagination by fear of mathematical equations which cannot be solved with reasonable effort. We have not yet found any problem that could not be reduced by circuit techniques to such a magnitude that a computer of reasonable size could handle it, if effects of minor importance are eliminated. The saving in time might compensate for the expenses of computer calculations in many instances.

It is sincerely hoped that this paper will instigate more widespread studies of air conditioning problems by application of circuit techniques. We have prepared a paper outlining in detail some of our studies.† The purpose will be to show how results of computer calculations utilizing circuit techniques can be readily used for practical everyday applications.

C. F. KAYAN, New York, N. Y.: This paper represents a further contribution to the studies of complex heat flow problems attached to building thermal load investigations. This latest and able formal presentation of the thermal circuit approach draws further attention to its status as a worthwhile tool, and deserves careful attention and cultivation.

The general procedure parallels the more specific and direct approach of the electrical analogy method of heat flow analyses in investigations, partially sponsored by the Society, described in various papers.△ Various phases of complex heat flow in both transient and steady-state analyses, have been studied by way of the analogous attack through electric circuits. Hydraulic methods have also been employed.**

The authors are to be applauded for their efforts to further popularize the circuit approach to building heat flow analysis. They are indeed welcomed with enthusiasm to the ranks of those others who have sought to add to calculation procedures for thermal problems by utilization of parallel procedures of other fields.

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‡ Cooling a Small Residence with a Two-Horsepower Mechanical Condensing Unit, by H. T. Gilkey, D. R. Bahnfleth and R. W. Roose (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 283).

† Analogue Computer Analysis of Residential Cooling Loads, by T. N. Willcox, C. T. Oergel, S. G. Reque, C. M. toeLaer and W. R. Brisken (A.S.H.V.E. TRANSACTIONS, Vol. 60, 1954, p. 505).

△ Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschkis (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 75). Electric Analogger Studies on Panels with Imbedded Tubes, by Carl F. Kayan (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 205). Electrical Analogger Application to the Heat Pump Process, by Carl F. Kayan (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 361).

** Hydraulic Analogue for the Solution of Problems of Thermal Storage, Radiation, Convection and Conduction, by C. S. Leopold (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 389).

E. P. PALMATIER, Syracuse, N. Y.: This paper is a very timely contribution to the technology of heating and ventilating. By reference to it, the estimator of cooling loads can construct all sorts of analogues, whose response characteristics to imposed loads may be investigated. It takes us step by step through a typical problem and indicates how the tedium of solution may be avoided by the application of modern computing equipment. Thus it gives us both understanding and courage to apply the method.

I have several questions I should like to submit to the authors as follows:

1. In the thermal circuits the mid-plane of the interior partitions, the floor and the ceiling, are shown to be at ground potential, which I assume is constant. The room air, which is connected directly to the ground through the load meter, also remains at ground potential. One would conclude, therefore, that the partition mid-plane is always at room air temperature. Obviously in an actual problem this probably is not the case. The mid-plane of the interior partitions must float up and down in a cyclic process the same as the imposed load, but not necessarily in phase with it. How does this affect the results or the solution to the problem? Can a correction be made by increasing the value of the capacitances C_b, C_c, C_a, etc?

2. Is the small peak in the sol-air input function that occurs about 6 a. m. the result of the Fourier series approximation of the input function, or does it actually exist in fact?

If the first is true, then should not some statement be made indicating that the corresponding little peaks or loops in Fig. 6 should be disregarded?

3. It would seem unnecessary for the authors or others to go to the trouble of computing complete solutions to the problem involving the radiant heat gains and other more complex problems. Would it not be sufficient to illustrate the setting up of the problem only?

Finally, it is important to state that such thermal circuits may be solved for constant values of the load current while allowing the temperatures of the room air, T_a, to follow a cyclic variation.

The authors are to be congratulated for introducing the circuit method to the heating and air conditioning engineer. Their work should stimulate practical application of the method.

P. R. ACHENBACH, Washington, D. C.: This paper presents a method for the determination of the instantaneous heat transfer to the indoor air resulting from cyclic changes in outdoor conditions under cooling operation. There is also an urgent need for information on the same type of problem under heating conditions. This need is more urgent now than formerly because the installation of unnecessary extra heating capacity in buildings employing electric resistance heaters or a heat pump is an economic burden both in first cost and in operating cost. The greater use of well-insulated components in building construction introduces greater lag and causes the heating load values computed for steady state conditions to be less representative of the actual heating needs.

Work is going forward in several Technical Advisory Committees that will in the future permit the engineer to make a logical choice of outdoor design temperatures that are suited to the heat capacity and lag characteristics of different types of construction. Parallel to this effort data should be obtained on the relation of the instantaneous maximum heating loads to the steady state heating loads for typical outdoor temperature and wind cycles of varying duration so the two pieces of information can be properly correlated. It appears to me that the cycle of outdoor temperature under heating conditions might often be represented by a simple sine curve as contrasted to the more irregular type shown in Fig. 4 of this paper, thus simplifying the calculating procedure considerably. The authors are to be complimented for this contribution toward the solution of an important and difficult problem.

P. E. MCNALL, JR., Minneapolis, Minn.: This paper shows that, even for the relatively simple case treated here, there are a lot of equations involved, even though they are not particularly difficult equations. That emphasizes the immensity of this problem as it faces the entire Society and others with regard to this subject.

It does appear to me, however, that fundamental approaches of this type are going to give a real contribution. For instance, the simple example given in the paper has already indicated some room for improvement in the sol-air temperatures and cooling load calculations given in THE GUIDE.

I think it should further be stressed that any method short of full-scale tests can only be as accurate as the physical data and the assumptions that are used in the solution. Our company is interested in circuit techniques; so much so that we are now constructing an analogue of a complete house and heating system, including a generalized control system for studies of this kind. Because the accuracy depends upon the physical data, this emphasizes the need more strongly than ever for better physical data, particularly for specific heats, densities, emissivities, and absorptivities of building materials.

In the discussion in the paper, the authors have briefly stated some of the advantages and disadvantages of the possible methods of approach to such problems. Analogue solutions have an important advantage particularly for those who are not mathematically inclined in that it is not even necessary to write the circuit equations, let alone solve them, when using an analogue of this kind. The results come out as direct physical measurements.

It should also be pointed out that all elements in a circuit except the sources and sinks are interacting. That is not generally the case in electronics and servo-analysis techniques which we are using for problems of this kind. A change in the geometry or the construction of any element will change all of the temperatures and heat flows in the circuit, many of them negligibly, that is true, but still they will change. This adds to the difficulty of drawing conclusions based on simple experiments without careful thought and study as to what they mean when generalized to other cases.

In this connection the inside glass temperature which the authors assume to be an independent variable is really a dependent variable. Their assumption is based on inside conditions comparable to those found in their previous test work at the A.S.H.V.E. Research Laboratory. For their case it is permissible to assume that it is an independent variable, but it is also affected by the temperatures of inside surfaces which exchange radiant heat with that window and the air velocities over it.

C. M. ASHLEY, Syracuse, N. Y.: There are still a number of steps which must be taken before the results of this study can be made practical for application to specific jobs. This is apparent in some of the assumptions which are made. One of the most important of these has to do with the question of the furnishings of the room which normally are of a low thermal mass and which influence greatly the effect of thermal storage in the system.

Another assumption is that of lumped rather than distributed areas of rooms. Both of these first assumptions are definitely on the unsafe side.

Of course, in addition, we need the solution of the distribution of the direct radiation from the window as well as for venetian blinds and shades and so forth. Also, the whole problem of the temperature distribution in the wall structure is one which it is very desirable to solve in a more exact manner. I believe that all of these things point to a rather long program but a very worthwhile one, from which we should be able eventually to get rather definite and specific results.

The practicing engineer, on the other hand, has overcome the obstacles by making an assumption in certain cases, particularly in residences, that the temperature will be allowed to vary within permissible limits, thereby permitting the assumption of a constant load. The results in practice have been excellent, but it is still desirable to have the confirming evidence which a theoretical analysis will give.

C. M. HUMPHREYS, Cleveland, Ohio: It might be well to point out that the authors of this paper are not suggesting that an engineer should write and solve all of the thermal circuit equations involved in his cooling system designs for an office or small store. If this were to be necessary, there would be very few cooling systems designed.

Instead, the thermal circuit is suggested as a means of evaluating the variables involved in the complex pattern of periodic heat flow which takes place in a heated or cooled structure. After these variables have been evaluated, and are understood, relatively simple approximations should permit much more accurate load calculations than we are able to make today.

One very simple application of the thermal circuit technique might be borrowed from Mr. Parmelee's discussion of housing in Israel. He mentioned the fact that while the typical houses of this area remain reasonably comfortable during the day, their temperature becomes almost unbearable at night. For maximum comfort at night or for optimum conditions during the 24-hr day, should these homes be of material having high or low heat capacity, and should the material be of high or low thermal conductivity? Should insulation be used, and if so, should it have high or low heat capacity, how much of it should be used, and where in the construction should it be located? We could probably get no agreement on the answers to these questions from

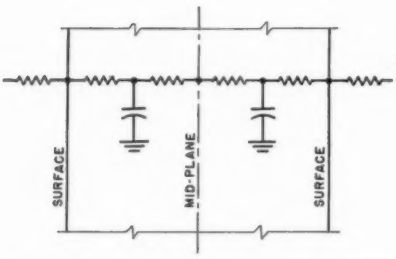


FIG. A. PARTITION CIRCUIT AS COMPLETELY VISUALIZED

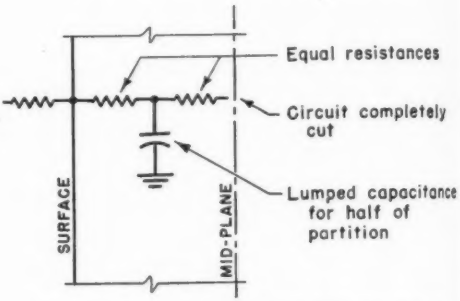


FIG. B. PARTITION CIRCUIT CUT SO THAT NO CURRENT WILL PASS THE MID-PLANE

the group of experts assembled here today, but the questions could be readily answered by the application of the thermal circuit technique.

AUTHORS' CLOSURE: It is stimulating to receive these well-considered discussions. Circuit analysis, in all of its broad applications, may be likened to a large set of tools. The engineer can make advantageous use of these tools in any combination which fits the problem defined, the nature of the results desired, the use which is to be made of the results, and the justifiable expenditure of time and constructive effort. Elaborate and specialized tools can be assembled to perform difficult tasks which could not otherwise be accomplished. A simple general-purpose tool can be selected to save time and effort. The engineer who enjoys exploring the what, how, and why of air conditioning system operation can draw upon varied assortments from the stock of tools, and with them study how things can be taken apart, put together, re-arranged, or adjusted by changing components.

Circuit techniques represent a broad domain wherein analogue techniques, as such, are but one part. Circuit techniques are a vitalizing embodiment of modern system methods of thinking and analyzing. System approaches consider all components and their interactions. Circuit diagrams are the symbolic language in which system problems are stated. Once stated, a problem is well on the road to solution. Obtaining the answer is an exercise in technique and judgment combined. Simplifications, approximations, deletions, substitutions,—all of these have their place.

Pausing in retrospect, circuit or system techniques were first discussed as the basis for a project at the A.S.H.V.E. Research Laboratory in 1946. Progress made in analysis of periodic heat flow, under the guidance of the Technical Advisory Committee on Cooling Load, and the development of more and better data on heat transfer through glass portions of structures, under the guidance of the Technical Advisory Committee on Heat Flow Through Glass, led to authorization for an initial exploratory study of circuit techniques.

A comprehensive analogue network for circuit studies was taken as a desirable ultimate objective. The method of the present paper, requiring only algebra and arithmetic, was developed for use as an inexpensive substitute for the rather heavy investment needed for an analogue. In 1953 an A.S.H.V.E. Cooperative Research Project to investigate and develop analogue methods was inaugurated at the University of California, Los Angeles.

The long-range goal in all of this is to make available, for convenient use throughout the engineering profession, general *solutions* to the most typical circuit problems.

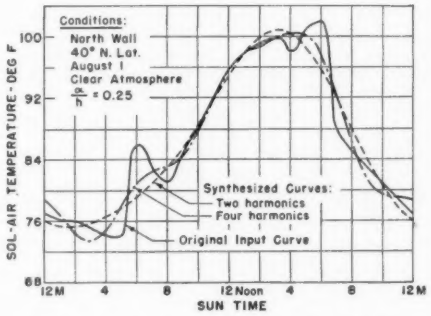


FIG. C. SOL-AIR TEMPERATURE CYCLE

The illustrative example in the present paper is merely an inadequate beginning. Cases for southern and western wall exposure, flat roofs, various proportions of glass, interior variations, furnishings, and a wide range of structural materials, are badly in need of attention. These problems *can be solved.*

It is very doubtful that a high degree of refinement in the thermal properties of the materials involved is necessary in solving problems of the above nature. Data on the time-variable characteristics of equipment are a pressing need, however, if the circuit is to be extended, as it should be, to encompass equipment. Equipment costs money, to buy, to install, and to operate; and savings are a strong incentive.

In the study of controls, circuit or system techniques are believed to be indispensable.

Looking ahead, we confidently predict that we are entering an era of system concepts, circuit techniques, and the ability to design, build, and operate from broadly integrating approaches. System methods bring out the total and the components in clear inter-relation.

We will be keenly interested in the results of Mr. Brisken's analogue investigations. We urge that he bring out complete details; including the design of circuit components.

Simplifications via circuit approximations certainly are in order, but we take the stand, during an initial *educational* phase, that it is best to start by drawing the complete circuit. Then, one has a basis for judging the very practical simplifications

which generally should be made for engineering purposes. An economic appraisal always is in order regarding expenditures *vs.* results in problems to be solved.

We do not wish to generalize concerning indoor radiation. Circuit methods, for example, can be applied to comfort problems wherein even small radiation currents can be significant.

With reference to the remarks by Professor Kayan, we feel that system concepts and circuit methods are *the* most specific and direct approach in analytical studies. Analogue techniques then are merely part of the broad approach. Moreover, system concepts are so unifying that the distinction between electrical, thermal, mechanical, hydraulic, magnetic, acoustic, and other problems disappears once the system has been diagrammed and defined.

We thank Mr. Palmatier for his perceptive encouragement. His questions are answered as follows:

1. *Mid-plane temperature for interior partitions:* A basic simplification was the idealization that partition mid-planes were planes across which no heat would flow at any time. This was our method for avoiding adjoining-space interactions. Circuit-wise, this means that the branch passing through the mid-plane was cut and left without any connection whatever. This is diagrammed in Figs. A and B and is also discussed in the appendices.

From Figs. A and B, it is seen that the mid-plane does not enter at all. The unconnected resistance in Fig. B is not a functioning part of the circuit. The potential on the circuit side of the capacitor refers to a point separated from the wall surface by one-quarter of the total partition resistance; this potential undergoes a cyclic variation.

The mid-plane temperature can be brought into the circuit by dividing the wall into an odd number of capacitance lumps. It may be easier, if the partition is solid, to obtain the mid-plane temperature from the surface-temperature variations by way of known analytical solutions.

2. *Peaks in input functions:* The peaks in question actually exist. They arise from direct solar incidence for small-time intervals on the north wall. Fig. C shows the actual and Fourier approximated sol-air temperatures.

3. *Set-up of other problems:* We do not follow this thought. The engineer must deal with quantitative results. The *more complex* problems may be far more important than the present simple case!

A circuit may be solved for any proper combination of imposed conditions and unknowns.

As pointed out by Mr. Achenbach, we recognize and accept the need for heating-system analyses. Very practical findings may be anticipated for heating problems. Reduced to primary essentials, the initial need is for some means of financing extensions of the thermal-circuit studies.

The aspect of required physical data mentioned by Mr. McNall is fully recognized. We recommend the circuit approach as a means for establishing *just how much refinement is really needed* in such data. Mr. Humphreys has brought this out very well in his discussion.

We caution users of circuit analysis against over-generalizations.

The glass temperature was introduced as a pre-determined known quantity as a simplification. There actually is a small error here, but this we chose to accept as being of entirely secondary significance for present purposes. This acceptance would *not* necessarily apply in all problems, whereby a circuit representation of the glass would be introduced. (This we have actually done on a trial basis.)

We agree most completely with Mr. Ashley and we emphasize that this paper is but a beginning. Mr. Humphreys has been very helpful in bringing out the aspects of the A.S.H.V.E. Research Laboratory's long-range plans for use of circuit techniques.

The important job remaining to be done is of far-reaching significance to the profession. The sincere interest shown here is an invaluable stimulation in obtaining support for continuation of the work.


1498

No. 1498

HEAT FLOW CHARACTERISTICS OF HOT WATER FLOOR PANELS

By E. L. SARTAIN* AND W. S. HARRIS**, URBANA, ILL.

DESCRIPTION OF EQUIPMENT

THE FLOOR SLAB LABORATORY, shown in Figs. 1 and 2, was constructed in 1947 for the specific purpose of studying methods of insulating unheated concrete floor slabs[1]. The one-story building was modified in 1949 and 1950 to accommodate concrete floor slabs with embedded warm-air perimeter heating ducts[2]. In 1951 the building was modified again to facilitate the investigation of hot-water panel heating using ¾-in. welded steel pipe embedded in concrete floor slabs and nominal ⅜-in. copper tube embedded in the plaster ceilings. Rooms A, B, and C were exposed only to the north while Room D had both a north and an east exposure; thus, the effect of solar radiation was minimized. Rooms A and B had one window each with an area equal to 10 percent of the floor area, while Rooms C and D each had two such windows. All windows were located in the north wall. A summary of the overall heat transfer coefficients and design heat losses for each room is given in Table 1. The walls between rooms were thoroughly insulated to minimize any heat transfer between adjoining spaces. The test room floors were separated by 6 in. of cellular glass insulation which extended from the floor surface into the ground to a depth of 4 ft. The insulation between the corridor and the test room floors extended to a depth of 2 ft.

The three methods of insulating the floor slabs studied in this investigation are shown in Fig. 3. The insulation used consisted of cellular glass, having a thermal conductivity of 0.42 Btu per (hr) (sq ft) (F) (in. of thickness). A 4-in. layer of coarse gravel under the floor provided suitable drainage and minimized the capillary rise of moisture from the soil. A duplex kraft paper was placed over the gravel fill for a membrane dampproofing. All the joints were lapped and then sealed with cold tar. The soil under and adjacent to the Laboratory contained 3 ft of fill which was primarily a silty clay composed of approximately 18 percent clay, 53 percent silt, 24 percent sand, and 5 percent gravel over approximately 1 ft of the original black silty clay top soil. The normal grade level around the building was 8 in. below the finished floor and sloped away from the building for proper drainage.

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[1] Temperature and Heat Loss Characteristics of Concrete Floors Laid on the Ground, by H. A. Bareither, A. N. Fleming, and B. E. Alberty (*American Artisan*, Vol. 87, No. 3, March 1950).

[2] Heat Emission Characteristics of Warm Air Perimeter Heating Ducts, by J. R. Jamieson, R. W. Roose, H. T. Gilkey, and S. Konzo (University of Illinois Engineering Experiment Station *Bulletin* No. 412).

Presented at the 60th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Houston, Tex., January 1954.

Heating System: A hot-water, floor panel heating system was installed in each test room. The panels were designed in accordance with the *I=B=R* Installation Guide No. 6 and consisted of ¾-in. welded steel pipe on 12-in. centers so located that the top of the pipe was 2 in. below the surface of the floor. In order to vary the output of the panel in conformity with the design heat loss of the room, various portions of the total coil length were by-passed by means of an arrangement of valves. The next to the last line of Table 1 shows the required panel area in each room based on a heat emission rate of 50 Btu per (hr) (sq ft). The last line gives the active panel area used during the tests reported in this paper. In each room hot water from the heater was supplied to the panel at the outside wall. Each test room was provided with its own



FIG. 1. FLOOR SLAB LABORATORY

electric water heater, watt-hour meter, circulating pump, necessary piping, and control system. A schematic diagram of the heating system is shown in Fig. 2.

Instrumentation: Complete instrumentation for temperature measurement was provided, consisting of approximately 600 copper constantan thermocouples which were used to measure the temperatures of room air, heated and unheated room surfaces, attic air, outdoor air, and water in the system. Along the center line of each room perpendicular to the north wall a large number of thermocouples were placed in the ground and on the under surface of the concrete slab. In addition, under and to the north of Room B, thermocouples were placed in the ground to a maximum depth of 20 ft (see Fig. 3).

Forty-four heat flow meters of the type constructed at the A.S.H.V.E. Research Laboratory[3], made and calibrated at the University of Illinois, were installed both on the top surface of the concrete floor slabs and under the gravel fill. Ten commercial heat flow meters were installed at the edges of the floor slabs in order to determine edge losses. Instruments were also provided to measure the relative humidity, electrical input to the immersion heaters and control circuits, water flow rates, and level of the water table under the build-

[3] A Low-Inertia Low-Resistance Heat Flow Meter, by R. G. Huebscher, L. F. Schutrum, and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 275).

ing. Continuous records of the more important temperatures and heat flow rates were obtained by the use of two recording potentiometers.

Auxiliary equipment included exhaust fans, heaters, and thermostatic controls for the attic, ventilating fans in the corridor, and thermostatically controlled convection heaters in the corridor and instrument room.

OPERATING CONDITIONS

For all of the tests discussed in this paper, each electric water heater was adjusted to a heat input rate equal to 1.3 times the estimated design heat loss of the room it served. This was to stimulate the normal piping and pickup

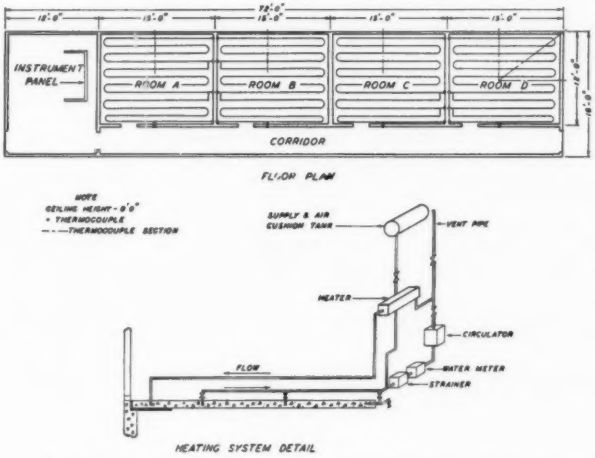


FIG. 2. FLOOR PLAN AND HEATING SYSTEM DETAIL

factor. The circulators were operated continuously, and the water heaters were turned on and off by means of thermostats in each room located 30 in. above the floor on an inside wall. During all tests reported in this paper, the thermostats were set to maintain a constant air temperature of 72 F at the 30-in. level. A high limit control was provided in each system and was adjusted to turn off the heater whenever the leaving water attained a temperature of approximately 130 F.

The corridor floor was maintained at the same temperature as that of the adjoining test room floor by manual regulation of the voltage of the supply to electric cables embedded in the corridor floor. The air in the corridor was kept at approximately the same temperature as that of the air in the test rooms by means of convection heaters thermostatically controlled. During all tests the temperature of the air in the attic was maintained at a constant value of 60 F by thermostatic control.

Complete thermocouple and meter readings for each room were taken every morning at approximately the same time, and the more important data were recorded over 24-hr periods.

RESULTS

Room Conditions: Table 2 shows the room-air and wall-surface temperatures as observed when the average outdoor temperature was 26.7 F. This outdoor temperature is about 10 F lower than the average for the heating season at Urbana, Ill., and so the data are representative of operating results for the greater part of the winter.

The air temperatures at the center of the rooms were very uniform, with a variation between the floor and 60-in. level of 0.5 deg. The temperatures of the air 3 in. below the ceiling and 3 in. above the floor were practically the same in all rooms except Room D where the difference was only 0.7 deg.

The inside surface temperature of the exposed wall in Room A was 2 to 3 deg lower than the room-air temperatures. The surface temperature of the

TABLE 1—HEAT LOSS DATA

(Based on Indoor Temperature of 70 F, Outdoor Temperature of −10 F)

ITEM	THERMAL CONDUCTANCE, BTU PER (HR) (SQ FT) (F)	DESIGN HEAT LOSS, BTU/HR			
		Room A	Room B	Room C	Room D
Surface					
Floor	0.69[a]	800	800	800	1,450
Walls	0.16	1,230	1,230	970	2,160
Ceiling	0.08	1,080	1,080	1,080	1,080
Windows	1.13	1,810	1,810	3,620	3,620
Infiltration	0.61[b]	1,100	1,100	1,100	1,100
Total Heat Loss		6,020	6,020	7,570	9,410
Panel Area Required, sq ft		120	120	151	188
Panel Area Used, sq ft		116	116	145	169

[a] Btu per hr per lin ft of exposed edge.
[b] Air changes per hr.

lower half of the east and west walls was about 2 deg warmer than the air, while the surface temperature of the south wall and the upper half of the east and west walls was about the same as the air temperature.

Table 2 shows that the floor panel system operating in a one-story basementless house produces warm floors and walls, and exceedingly uniform room-air temperatures.

Heat Flow from Panel vs. Energy Input to Water: Heat flows from a heated floor slab in three directions: upward into the room, downward to the ground or crawl space, and horizontally from the exposed edge to the outdoors. Heat meters were installed on the top surface, along the exposed edge and under each floor slab, in positions where the heat flow rates were believed to be representative of those for the total respective areas (Fig. 3). This being true, the total heat flow in any direction could be determined by multiplying the respective area by the average heat flow rate in that direction as determined by the heat flow meter. The total flow from the panel was the sum of the total flows in each of the three directions.

The energy input to the water from the heater and the pump motor was equal to the total heat output of the panel plus the heat loss of the connecting

piping. Water temperatures were taken at the inlet and outlet of the heater and at the inlet and outlet connection to the panel. From these observations it was found that the heat loss from the insulated connecting piping was negligible. The heat loss through the walls of the heater could not be measured, but since the heater consisted of about 2 ft of well insulated 2-in. pipe and average water temperatures were ordinarily less than 130 F, this heat loss would not be large in comparison to the total panel output. Therefore, the total energy input to the water was also a good measurement of the total heat output of the panel.

Over a wide range of outdoor temperatures, the total output of the floor panel in each room was determined from the heat meter readings and the total energy input was obtained from the readings of the watt-hour meters in the heater and pump motor circuits. It was assumed that 60 percent of the energy input to the pump motor was utilized in raising the water temperature. The relationship of these two measurements is given in Fig. 4. Table 3 summarizes the observations made in each of the four test rooms.

Except for Room C the two methods of measuring the total heat output of the panel agreed within 10 percent.

Because of the lack of test points at high indoor-outdoor temperature differences, the curves for all rooms were extrapolated to design conditions. This extrapolation could account for part of the discrepancies at high indoor-outdoor temperature differences. One heat meter in Room C was located on the boundary of the active and inactive panel areas. This condition did not exist in other rooms. The heat output rate changed rapidly along this line and it is quite probable that the readings of this meter were not representative of the heat

TABLE 2—ROOM-AIR AND WALL-SURFACE TEMPERATURES

(Average Outdoor Temperature 26.7 F)

DISTANCE ABOVE FLOOR INCHES	ROOM-AIR TEMPERATURES (F) CENTER OF ROOM			
	Room A	Room B	Room C	Room D
3	71.2	71.9	72.3	72.5
12	71.2	71.8	72.6	72.6
30	71.2	71.9	72.8	73.0
60	71.6	72.1	72.3	73.1
84	71.7	72.2	72.3	73.2
93	71.0	72.0	72.3	73.2

DISTANCE ABOVE FLOOR INCHES	WALL SURFACE TEMPERATURES (F) —ROOM A			
	South Wall	North Wall	East Wall	West Wall
3	71.8	70.0	74.2	74.8
12	72.1	69.5	74.2	74.2
30	72.2	69.2	73.9	73.4
60	72.2	68.8	72.5	72.8
84	72.2	68.8	71.7	72.2
93	72.2	68.2	71.8	71.8

Note: Air temperatures obtained at the various levels 2 ft from the thermostat slightly higher than those obtained at center of the room.

flow from the active panel area. The heat flow rate along this line was probably higher than that indicated by the heat meter. Had the meter been located within the active panel area in Room C, the heat meter and electric meter data would have been in better agreement.

In all the rooms, the heat flow as measured by the heat meters was consistently lower than the heat equivalent of the energy input to the electric heaters and work done by the pumps. A slight difference in this direction was to be expected since the energy input to the heaters included the small loss from the heater wall and from the piping between heater and panel. On the other hand, the

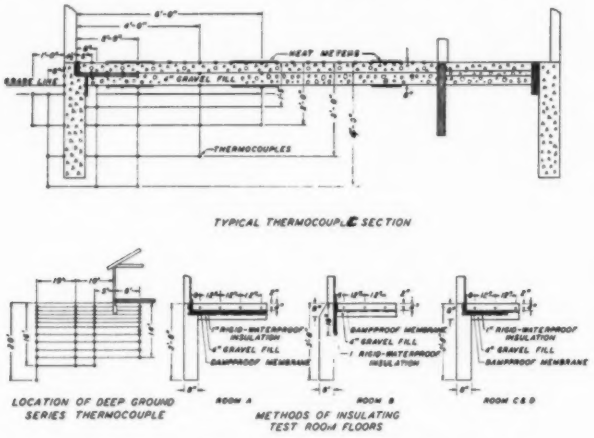


FIG. 3. INSTRUMENTATION AND FLOOR INSULATION

heat meter readings of necessity covered only a small percentage of the total panel area. The assumption that the heat flow rate determined from this small sampling was representative of the total heat flow rate of the panel could result in inaccuracies. The fact that in practically every instance the agreement was within 10 percent for the two methods of measurement was considered as evidence that instrumentation was adequate. Since there appeared to be fewer chances of error in the measurements of heat flow based on the electrical input to the system, this method of measurement was used for the total heat flow, and the heat meter data were used to proportion the heat flow from the panel in each direction.

Distribution of Heat Flow from Panels: Fig. 5 shows the total heat flow from the panel of each room as measured by the electric meters in the pump motor and heater circuits. Fig. 6 shows the percentage distribution of the total heat flow in each direction from each of the panels as measured by the heat meters. These curves are all based on conditions of operation by which the rooms were maintained at a constant temperature of approximately 72 F at the 30-in. level at all times.

From Fig. 6 it may be observed that for all indoor-outdoor temperature differences in excess of 30 deg the division of heat flow from the panels in all

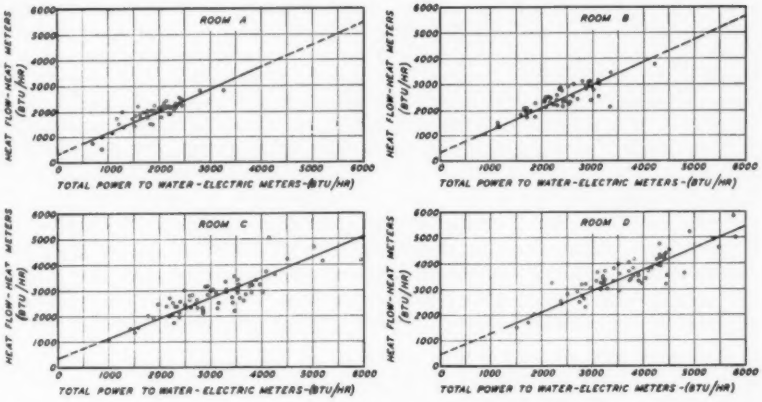


FIG. 4. BTU OUTPUT OF PANELS — ELECTRIC METERS *vs.* HEAT METER MEASUREMENTS

four rooms was about the same, with the useful heat into the room being 3 to 5 percent higher in Room A than in the other rooms. At indoor-outdoor temperature differences of less than 30 deg there were wide differences in the division of the heat flow from the different panels, with Rooms A and B showing a greater proportion of the total heat being supplied to the rooms.

Actual heat flow in any direction from a panel may be obtained by multiplying the total heat flow at any given indoor-outdoor temperature difference as read from Fig. 5 by the corresponding percentage value read from Fig. 6. Table 4 has been constructed in this manner for a temperature difference of 80 F in order to compare the measured heat flow into the rooms with the calculated heat losses. The total measured heat flow from the panels ranged from 90 percent of the total calculated heat loss of Room A to 111 percent of the calculated loss of Room C. The measured upward heat flow from the panels ranged from 87 percent of the calculated above floor heat loss in Room A to 101 percent of the calculated above floor heat loss in Room C.

It should be pointed out that the constructions of Rooms A and B were supposedly the same, and therefore the calculated heat losses were the same; however, the measured upward heat flow from Panel B was about 13 percent higher than that of Panel A. Later investigation revealed some torn insulation in the wall of Room B and one small wall area with no insulation. This was repaired after which the measured heat inputs to Rooms A and B were in better agreement. All of the tests reported in the paper were made prior to the time the wall insulation in Room B was repaired. While the condition of the insulation had some effect on results obtained, as pointed out in subsequent sections of this paper, the effect was not of sufficient magnitude to alter conclusions drawn.

While Fig. 6 indicated that there was a large percentage difference in the heat flow downward and from the edge of the different panels at low indoor-outdoor temperature differences, the difference in absolute values was small.

In order to make a true comparison of the effectiveness of the insulation, the data had to be integrated over the whole range of outdoor temperatures experienced. The first two columns of Table 5 show the average outdoor temperature conditions experienced in Urbana, Ill., from 1936 to 1949 for the months of January, February, March, April, May, September, October, November and December. The rest of the table indicates the total reverse losses on a yearly basis based on Figs. 5 and 6. These values were then totaled and the differences taken. The reverse loss of the panel in Room B exceeded that of Room A by 623,120 Btu per year, while the reverse loss in Room C exceeded that of Room A by 1,609,930 Btu per year. Assuming a seasonal operating efficiency of 70 percent, these increases in the reverse losses represent yearly increases in the seasonal fuel consumption of 8.9 and 23.0 therms of gas, or 6.4 and 16.4 gal of fuel oil (140,000 Btu per gal), respectively. This small saving obviously does not warrant the additional expense of providing insulation under the entire slab.

TABLE 3—COMPARISON OF PANEL OUTPUTS AS MEASURED BY HEAT METERS AND ELECTRIC METERS

Indoor-Outdoor Temp Diff F Deg	Room A			Room B			Room C			Room D		
	Panel Output			Panel Output			Panel Output			Panel Output		
	Watt-Hr Meter Btu/hr	Heat Meter Btu/hr	% Diff[b]	Watt-Hr Meter Btu/hr	Heat Meter Btu/hr	% Diff[b]	Watt-Hr Meter Btu/hr	Heat Meter Btu/hr	% Diff[b]	Watt-Hr Meter Btu/hr	Heat Meter Btu/hr	% Diff[b]
20	1130	1250	1.1	1330	1460	9.8	1450	1410	− 2.8	1800	1980	10.0
40	2280	2250	−0.3	2770	2790	0.7	3450	3050	−11.7	3980	3810	− 4.3
60	3420	3200	−6.4	4180	4060	−2.9	5400	4660	−13.7	6120	5600	− 8.5
80	4540[a]	4220[a]	−7.1	5580[a]	5350[a]	−4.1	7400[a]	6300[a]	−14.9	8350[a]	7500[a]	−10.2

a Extrapolated values.
b Based on electric meter data.

In order to make fair comparisons of subfloor losses for the two types of edge insulation at the same indoor-outdoor temperature differentials, the following conditions should exist:

1. The average water temperature in the panels should be equal.
2. The panel areas should be equal.
3. The above floor room characteristics should be the same.
4. The ground conditions should be the same.

The same design water temperature was used in each room and therefore for any operating condition the water temperatures were approximately the same for each room. Ground conditions were as uniform as it was possible to make them, and since normal variations in the film coefficient on the room side of the panel would produce only a very small change in the downward heat flow, only the difference in the panel area could have an appreciable effect upon its magnitude.

The panel area in Room C exceeded that used in Room B by 29 sq ft. This additional area was located on the side of the room opposite the exposed edge. A test at an average outdoor temperature of 26 F indicated that the increase in downward heat flow amounted to 1.9 Btu per hr for each sq ft of additional panel area. Assuming this to be a representative average value for a heating

TABLE 4—UPWARD HEAT FLOW FROM PANELS COMPARED WITH CALCULATED HEAT LOSSES

ITEM	ROOM A	ROOM B	ROOM C	ROOM D
Calculated Above Floor Loss (Unheated Attic) (Btu/hr).	5,200	5,200	6,770	7,950
Calculated Above Floor Loss (Heated Attic) (Btu/hr)...	4,300	4,300	5,850	7,040
Calculated Total Loss (Unheated Attic) (Btu/hr).....	6,020	6,020	7,570	9,410
Calculated Total Loss (Heated Attic) (Btu/hr)	5,100	5,100	6,650	8,490
Total Heat Flow From Panel (Fig. 5).....	4,580	5,600	7,400	8,350
Upward Heat Flow in Percent of Total (Fig. 6).....	82	76	79.5	78.5
Measured Upward Heat Flow.....	3,760	4,260	5,880	6,550
Ratio of Total Heat Flow from Panel to Total Calculated Loss (Heated Attic).....	0.90	1.10	1.11	0.99
Ratio of Upward Heat Flow to Above Floor Loss.....	0.87	0.99	1.01	0.93

Note: Calculated Losses based on: Indoor temperature = 70 F; Outdoor temperature = −10 F; Heated attic temperature = 60 F; Unheated attic temperature = −10 F.

season consisting of 246 days, the downward heat flow from the panel in Room C corrected to the panel area used in Room B would be $3{,}987{,}410 - (1.9 \times 24 \times 29 \times 246) = 3{,}662{,}100$ Btu per heating season as compared to 3,000,600 Btu per heating season for Room B. Thus, the C-type insulation appears to be about $3{,}000{,}600 \div 3{,}662{,}100$, or 82 percent as effective as the B-type insulation in reducing downward heat flow. While percentage-wise the C-type insulation is only 82 percent as effective as the B-type, the difference in the seasonal reverse loss was 3,662,100 — 3,000,600 or 661,500 Btu. At 70 percent efficiency

TABLE 5—SEASONAL REVERSE LOSSES FROM PANELS[a]

(Edge and to Ground)

AVG OUTDOOR TEMP (F)	AVG NO. DAYS PER YEAR	TYPE A		TYPE B		TYPE C	
		Subfloor Loss Btu/day	Total Loss Btu/year	Subfloor Loss Btu/day	Total Loss Btu/year	Subfloor Loss Btu/day	Total Loss Btu/year
−10 to − 5	0.2	18,900	3,780	30,300	6,050	35,700	7,130
− 5 to 0	0.4	18,130	7,250	28,400	11,350	34,200	13,680
0 to 5	0.8	17,200	13,750	27,000	21,600	32,100	25,700
5 to 10	2.2	16,420	36,200	24,900	54,800	30,500	67,200
10 to 15	4.6	15,550	71,500	23,100	106,300	28,500	131,200
15 to 20	7.6	14,900	113,200	21,450	163,000	26,400	202,000
20 to 25	13.6	14,020	190,800	19,950	271,500	24,500	333,500
25 to 30	25.4	13,100	333,000	17,900	455,000	22,800	579,000
30 to 35	33.8	12,250	414,000	16,150	545,000	20,700	700,000
35 to 40	30.0	11,410	343,000	14,400	432,000	18,900	567,000
40 to 45	23.4	10,200	238,000	12,150	284,000	16,800	393,000
45 to 50	22.6	9,040	204,000	10,600	239,000	15,400	348,000
50 to 55	20.8	7,870	163,000	8,400	175,000	12,700	264,000
55 to 60	19.8	6,620	131,000	6,720	133,000	10,300	204,000
60 to 65	22.0	5,220	115,000	4,680	103,000	6,930	152,000
65 to 70	19.0	3,130		2,100		4,800	
Totals.....	246.2		2,377,480		3,000,600		3,987,410
Difference from Room A.					623,120		1,609,930

[a] Based on Electric Meter Data.

this is equivalent to 9.2 therms of gas or 6.7 gal of oil. Obviously this difference is of no economic value.

Ground and Edge Losses from a Heated Floor Slab: Fig. 6 shows that at design conditions of 80 F indoor-outdoor temperature difference, the downward heat flow in Rooms B and C amounted to 19.0 and 16.5 percent of the total heat output of the panels, respectively. The heat loss at the edge of both panels was 4 percent of the total. Thus, 23.0 percent of the total output of the panel in Room B and 20.5 percent in Room C was lost directly to the ground or to

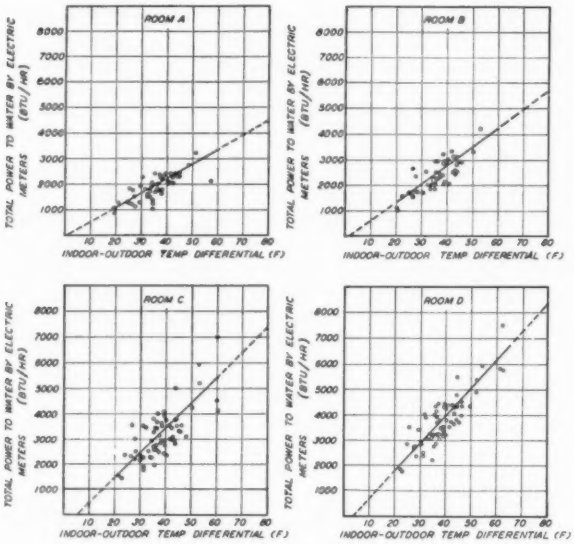
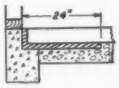
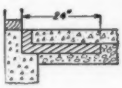
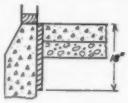
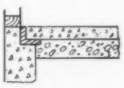


FIG. 5. TOTAL HEAT OUTPUT OF PANELS

the outdoors without being of value to offset the above floor heat losses of the rooms. This is commonly referred to as the reverse loss from the panel.

At the design condition the total heat flow from the panels as determined from Fig. 5 was 5,600 Btu per hr in Room B, and 7,400 Btu per hr in Room C. The reverse losses were 23.0 and 20.5 percent of these values, or 1,290 and 1,515 Btu per hr respectively. After correcting for panel area as described in the preceding section, the reverse losses were 1,340 and 1,515 Btu per hr, respectively, based on the panel area of Room C. These reverse losses are equivalent to 93 Btu per hr per linear ft of exposed edge for Room B, and 104 Btu per hr per linear ft of edge in Room C. Table 6 shows the heat loss, expressed in Btu per hr per linear foot of exposed edge, from both unheated slabs and concrete floor panels of various types of construction. The losses from the unheated floor slabs were obtained from the HEATING VENTILATING AIR CONDITIONING GUIDE. A comparison of the 1-in. vertical type insulation shown in Column C indicates that the reverse loss for the concrete floor panel is 43 Btu

TABLE 6—HEAT LOSSES OF HEATED AND UNHEATED FLOOR SLABS
(Btu per linear foot)

FLOOR SLAB	 A	 B	 C	 D
Unheated[a]	50[b]	45	50	55
Heated	104		93	
Diff	54		43	

a THE GUIDE 1953, pp. 232-253.
b Estimated.

per hr per linear ft of exposed edge in excess of that for the unheated slab. As applied to Room C, this would amount to 43×14.5 or 623 Btu per hr total reverse loss over that for the unheated slab. Thus, using vertical insulation and at a design condition of 80 F indoor-outdoor temperature difference, the fuel required to heat Room C by a concrete floor panel system would be $623/(7400 - 623) \times 100$ or 9.2 percent greater than that required to heat with a more conventional system. The loss for the unheated slab using the 1-in. edge insulation shown in Column A, Table 6, would be somewhere between 45 and 55 Btu per linear ft because it would be better than the 1-in. edge insulation shown in Column D and not as good as the 2-in. edge insulation shown in Column B. Using a value of 50 and comparing it with 104 for a concrete floor panel with edge insulation shown in Column A, the fuel required to heat Room C at design conditions by a concrete floor panel system would be $783/(7400 - 783) \times 100$ or 11.8 percent greater than that required to heat with a more conventional system utilizing the same edge insulation.

Floor Surface Temperatures: Floor surface temperatures were obtained by means of thermocouples installed on 6-in. centers on a line perpendicular to the north wall and passing through the center of the room. These thermocouples were so arranged that they were located alternately over and midway between coils.

Fig. 7 shows a plot of the heat flows and floor surface temperatures for each of the test rooms at a time when the outdoor temperature was 26.7 F. The

TABLE 7—EFFECT OF RADIATION AND CONVECTION ON HEAT FLOW FROM PANEL

CONDITION	BTU PER HR (SQ FT)		
	Meter U-24	Meter V-17	Average
Heat Meters Exposed to Window	25.8	26.3	26.0
Baffle between Heat Meters and Window 8 in. Clearance between Floor and Baffle	21.9	21.4	21.6
Baffle between Heat Meters and Window No Clearance between Floor and Baffle	15.5	16.8	16.1

figure shows that the maximum variation in floor surface temperature between adjoining active tubes was about 3 deg, while the average surface temperature across the active panel remained practically constant. Even though there was little change in the floor surface temperature across the active panel area, the upward heat flow as measured by the heat meters varied from about 15 Btu per hr per sq ft at the center of the room to about 25 Btu per hr per sq ft near the window. To determine how much of this increase might be assigned to increased radiation and how much to increased convection due to the presence of the window, a piece of insulation board 4 ft high by 8 ft long was placed in a vertical position between the windows in Room D and those heat

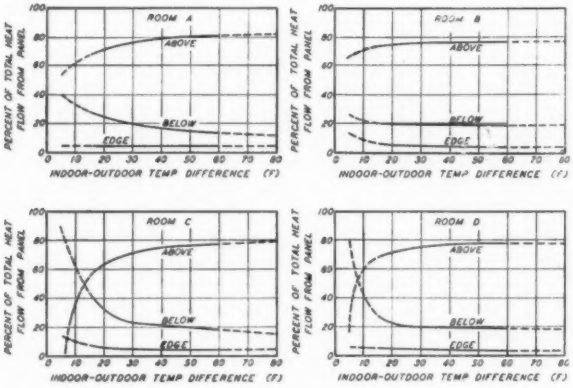


FIG. 6. PERCENTAGE DISTRIBUTION OF TOTAL PANEL OUTPUT

meters nearest to the outer wall. With this baffle located 8 in. above the floor, it offered very little resistance to normal convection currents over the floor, but it did shield the heat meters from the radiation effects of the windows and most of the exposed wall. Dropping the baffle to the floor not only shielded against radiation effects, but also prevented the circulation of cool air from the window and exposed wall over the floor in the area of the heat meter location.

The effect of the baffle location on the heat meter readings is shown in Table 7. When the heat meters were exposed to the window (no baffle) the average upward heat flow rate was 26 Btu per hr per sq ft. With 8 in. clearance between the bottom of the baffle and the floor, the average upward heat flow was reduced to 21.6 Btu per hr per sq ft, and with the baffle located in contact with the floor, the upward heat flow was reduced to 16.1 Btu per hr per sq ft. In other words, increased radiation to windows and cold wall accounted for about 45 percent of the total increase in upward heat flow from the panel near the windows and convection currents accounted for about 55 percent of the total increase.

These readings indicate that a floor panel system has the desirable characteristic of automatically increasing the heat output rates in areas adjacent to

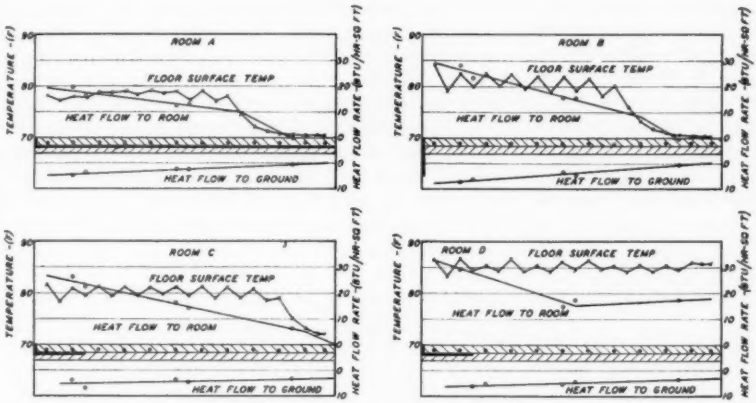


FIG. 7. HEAT FLOW RATES AND PANEL SURFACE TEMPERATURES

points of high heat loss from the room. For a given floor panel surface temperature and room-air temperature, a somewhat higher output per square foot of panel area could be expected in an uninsulated room or in a room with large glass area than in a fully insulated room with limited glass area.

The average effective panel surface temperature for each room was obtained by averaging the readings of all floor surface couples located on the heated portion of the floor. For each room this average temperature was plotted against the indoor-outdoor temperature difference corresponding to the time the readings were taken (Fig. 8). The maximum difference in the average panel surface temperature for all rooms was about 3 deg at any indoor-outdoor

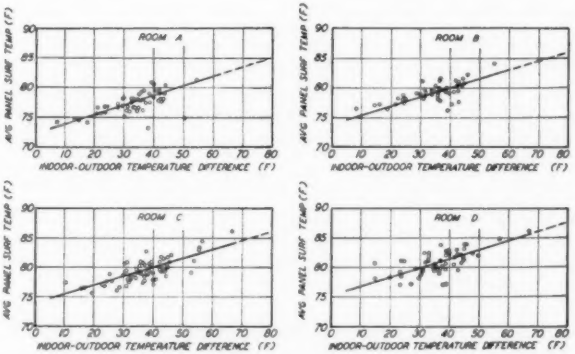


FIG. 8. RELATIONSHIP BETWEEN INDOOR-OUTDOOR TEMPERATURE DIFFERENCE AND PANEL SURFACE TEMPERATURE

temperature difference. Some difference would be expected since the installed panel area was based on the calculated heat loss of the room, and the ratio of the actual heat loss to the calculated ranged from 0.90 for Room A to 1.11 for Room C. It is also true that the panel surface temperatures as measured on the north-south center line of the rooms were not necessarily the same as at other points in the rooms. This was especially true in Rooms A and B where the effect of the smaller window area on the floor temperatures was greater along the center line of the rooms than toward the sides of the rooms. On the other hand, the larger window areas in Rooms C and D affected a greater percentage of the total panel area and hence the temperatures measured along

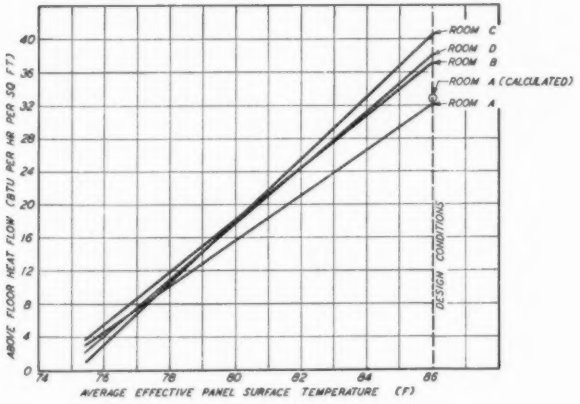


FIG. 9. PANEL OUTPUTS TO ROOMS

the center line in these rooms probably more closely approximated the true average surface temperature.

It is probable that had the instrumentation in each room been adequate to obtain the true average panel surface temperature for the entire panel area, the difference between rooms would have been even less than that indicated in Fig. 8.

The panel surface temperatures as measured in Rooms B and C were accepted as being representative of the true average for each of the four test rooms, in that the average panel surface temperature as measured on the north-south center line of a room was not exactly representative of the true average, and the curves for Rooms B and C were the mean of all four rooms with only a 1.5 deg deviation from the extremes.

The curves of Fig. 9 were constructed by dividing the upward heat flow from each panel (Figs. 5 and 6) at any given indoor-outdoor temperature difference by the panel area and plotting the quotient against the corresponding panel surface temperature as obtained from the mean of the curves in Fig. 8.

These curves show the relationship between panel output to the test room and panel surface temperature. At a design condition of 80 deg indoor-outdoor temperature difference, the average panel surface temperature for all rooms was taken as 86 F, while the panel output varied from 32.4 Btu per hr per sq ft for Room A to 40.6 Btu per hr per sq ft for Room C.

The output of the panel in Room B was higher than that of Room A, while the calculated heat losses of the two rooms were the same. Due to faulty insulation, the actual heat loss of Room B was higher than that of Room A. Since the two rooms had the same panel area, it follows that the heat emission rate from Panel B must have been higher than that from Panel A, either as a result of higher operating temperature or of an increase in radiation because of the lower wall surface temperature over the poorly insulated area. Had the insulation in Room B been identical to that in Room A, the curve for Room B in Fig. 9 would have approached that for Room A.

It can be seen that at design conditions, the output of the panels in Rooms C and D, with two windows each, was 20 to 25 percent greater than the output of the panel in Room A, with only one window. This is in agreement with the results of the special test showing the effect of windows on panel output.

The circle plotted at design conditions in Fig. 9 was obtained by calculation using the formulae 3 and 5 given on pages 555 and 556 in THE GUIDE 1953 substituting an average unheated surface temperature of 66.5 F, a room-air temperature of 72 F, and a panel surface temperature of 86 F. All these values correspond to observed conditions in Room A at an indoor-outdoor temperature difference of 80 F. It can be seen that the output of the panel in Room A was essentially the same as the calculated value while the panel outputs in Rooms C and D where higher than the calculated value by almost 20 percent, or about 6.5 Btu per hr per sq ft. Had the actual average unheated surface temperatures for Rooms C and D been known and used in calculating the panel output, the calculated output would have more nearly approached the test values for these rooms.

CONCLUSIONS

The following is a summary of the results obtained for the test conditions investigated.

1. The air temperatures at the center of the rooms were very uniform, with a variation between the floor and 60-in. level of 0.5 deg. The temperatures of the air 3 in. below the ceiling and 3 in. above the floor were practically the same.

2. The measured heat flow from panel to room ranged from 87 percent of the calculated above floor heat loss in Room A to 101 percent of the calculated above floor loss in Room C.

3. It was found that the fuel savings resulting from the use of insulation under the entire floor slab as compared to the use of edge insulation only was too small to warrant the additional cost.

4. Vertical insulation along the inside edge of the foundation wall was as effective as the L type edge insulation. The savings in material and the ease of installation made the vertical insulation the more desirable of the two types.

5. At design conditions, the reverse loss from panels with edge insulation amounted to 20 to 23 percent of the total panel output. The reverse loss was roughly twice as great as the estimated heat loss through unheated floor slabs using the heat transmission values given in THE GUIDE 1953.

6. Comparisons of the reverse loss from the heated slabs, with figures presented in Chapter 12 of THE GUIDE 1953 for heat losses from unheated floor slabs, indicate that at an indoor-outdoor temperature difference of 80 F, the fuel consumption when using floor panels, would exceed the fuel consumption when using radiators or convectors by about 10 percent.

7. It was found that while the mean floor surface temperature was uniform across the panel, the heat emission rate was much higher near the exposed wall and window than at the center of the room. Thus, the floor panel system had the desirable characteristic of automatically increasing the heat output rate in areas adjacent to points of high heat loss from the room. For a given floor surface temperature and room-air temperature, one could expect a somewhat higher output per square foot of panel area in an uninsulated room or one with large glass area than in a fully insulated room with limited glass area.

8. At design conditions, the measured heat transfer from the panel to Room A was 1.5 percent less than the calculated output obtained by substituting actual measured air, panel surface and average unheated surface temperatures in Equations Nos. 3 and 5 in Chapter 24 of THE GUIDE 1953.

ACKNOWLEDGMENT

The authors wish to acknowledge the assistance of the Small Homes Council, a division of the University, which made available the laboratory and many of the instruments used in the tests. The authors also thank those companies which have supplied equipment and instruments.

DISCUSSION

P. R. ACHENBACH, Washington, D. C.: The authors have provided information through this investigation that will be very useful to the designers of floor panel heating systems and to the home building industry.

I was not able to discern the location of the commercial heat flow meters on the slab edges in Fig. 3. I would not expect that vertical heat flow meters would provide an accurate measure of the heat flow through the edge of the slab because the direction of heat flow would not be exactly perpendicular to the meters and therefore considerable judgment would be required in assigning the proper area to the heat path and the proper correction to the observed temperature differences.

Was the relatively good agreement between computed heat flow from the panel based on heat flow meters and that computed from the electric meters, as shown in Table 3 of the paper, a fortunate circumstance or do you think the method is consistently susceptible of such good agreement? Might not the greater disparity between computed panel outputs and electrically-measured outputs at lower outdoor temperatures be caused by a lag in the cooling of the earth under slab? The heating panel and the gravel and earth beneath it probably were nearer to steady state during mild weather than during cold spells, which often do not last long.

Referring to Fig. 7 of the paper, it would not be expected that the heat transfer from the floor surface would be nearly twice as great near the window as at the center of the room unless there was a greater temperature difference between the embedded pipe and the floor surface near the edge. It is believed that the direction of flow of water in the piping system assists in the explanation of the observed results which illustrate the desirability of having the hot water enter the panel near the cold wall.

The results in Table 7 provide information on the effects of radiation and convection on the upward heat transfer from the floor near the cold wall. The effect of radiation may be somewhat greater than is indicated by these data because placing a vertical baffle near a window that does not touch the floor would increase the down-

ward chimney effect at the window and deliver more cold air to the floor surface than would take place without the baffle. We have seen this demonstrated with full-length drapes over large windows in a number of cases.

Why was 1 in. of insulation used for the present tests of heated floors when previous investigations at the University of Illinois with heated slabs strongly indicated that a 2-in. thickness was preferable?

R. G. VANDERWEIL, Boston, Mass.: This paper has suggested three relatively minor questions to me.

In Fig. 4 I noted that heat output was indicated from the panel without indication of heat input to the panel. That obviously should have been caused by some non-steady flow condition, heat inertia of the ground or some other secondary heat flow.

I also noticed that the authors had located a number of thermocouples between 12 and 20 ft below the panel in the ground. Hence, I wonder whether any attempt was made to measure possible reverse flow from the ground to the panel.

The second question is related to the first. Fig. 6 shows the downward heat loss to be essentially constant, with very small temperature differences between outside and rooms. There again I assume the cause is to be found in the fact that the differential to the soil temperature below has remained fairly constant.

A third question is the indication in the conclusion of this paper that for the construction shown, panel losses of somewhere between 15 and 25 percent are normal. I wondered whether such losses would not seriously increase where buildings are located on moist soil, particularly where ground water might be near the surface and in a flowing state.

P. E. MCNALL, JR., Minneapolis, Minn.: The authors showed that the heat flow rate from the surface of the panel near the outside wall and windows was greater than that near the inside walls although the surface temperature showed no increase near the outside walls. Another discussor pointed out the hotter water at the outside increases the temperature drop between the pipe and the panel surface. That certainly is one effect to be considered here. I inferred from the authors' statement that this type of increased heat flow from the panel near that outside wall was to be desired. I disagree because if that happened to the panel at the same surface temperature it might happen to me if I were sitting near that outside wall and window. My heat loss would also increase. I think that for comfort, the panel surface temperature at that outside wall should be increased more than is shown in this test.

R. S. DILL, Washington, D. C.: I would like to know if it rained during the test and what effect dampness would have on the heat transfer.

P. B. GORDON, New York, N. Y.: Table 4 and Fig. 5 present information on the total heat output of the panels. It would appear from Table 4 that the measured heat output for the 80 deg temperature difference, as compared with the expected or calculated output, is generally less.

I note that this is developed by extending the data in Fig. 5, the A, B, C, and D lines out to the 80 deg temperature difference. Is it valid to use this value as an indication of a measured output at the 80 deg temperature difference when extended as a straight line, or is it possible that that straight line may slope up as the temperature difference becomes greater? For example, at the 60, 70 or 80 deg difference, because the influence of infiltration becomes stronger, is it not possible that the total output may be closer to the calculated output?

AUTHORS' CLOSURE (W. S. HARRIS): In reply to the questions raised by Mr. Achenbach; two heat meters were located at the edge of the slab in each room—one opposite the 4-in. edge of the slab on the cold side of the insulation and the other along the inside edge of the foundation wall directly under the 4-in. concrete floor slab.

It is true that the percentage error in the measurement of edge losses may have been large. However, expressed in Btu per hour these errors of measurement were small as compared to the total heat flow from the panel and therefore did not have an appreciable effect on the percentage distribution of total panel output (Fig. 6).

Air velocities were not measured when obtaining the data presented in Table 7 so we have no information on the effect of the baffle on the air velocities over the floor when the bottom of the baffle was 8 in. above the floor. The baffle in this position reduced the upward heat flow from the panel at heat meters U-24 and V-17 by 4.4 Btu per hr. If air flow was increased by the baffle then the reduction in radiant output must have been somewhat greater than 4.4 Btu per hr.

One inch of insulation was used in that this is the amount many building contractors are using even though 2 in. has been recommended.

Mr. Dill inquired about the effect on heat transfer rates of excess moisture in the ground from rain or snow. Weather conditions during tests were recorded but at this time no effort has been made to correlate downward heat flow rates with the moisture content of the soil.

The authors agree with Mr. McNall that it is desirable to increase the inside surface temperature of walls and windows. By the use of insulation the inside surface temperature of walls can be raised to within 3 or 4 deg of the indoor air temperature when it is zero outdoors. On the other hand not too much can be done to make the inside surface temperature of windows warmer. When it is zero outdoors and 70 F indoors the inside surface temperature of single glass will be about 18 F and with storm sash 41 F. These surface temperatures are sufficiently low that without sufficient heat supply to the room under the glass area they may be a possible source of cool convection currents across the floor. The increase in panel output near the windows helped to insure the elimination of drafts. If the panel surface temperature along exposed walls and especially near windows can be maintained a little higher than for the rest of the floor panel this will provide additional protection against drafts originating at windows or cold wall surfaces.

Mr. Gordon questioned the validity of a straight line extrapolation of the curves of Fig. 5 to an 80 deg indoor-outdoor temperature difference. The extrapolation was necessary because of lack of outdoor temperatures as low as −10 F during the tests. It is possible that had sufficient data been obtained with colder outdoor temperatures the lines in Fig. 5 would curve slightly. On the other hand, similar data have been obtained at the $I=B=R$ Research Home and the Warm Air Research Residences over a number of years and for the most part these data indicate the validity of straight line extrapolation of the curves in Fig. 5.

The study of heat flow between the panel and the ground is being continued and the authors expect to have more complete answers to Mr. Vanderweil's questions when these studies are completed. We do know some of the heat flowing from the panel to the ground during cold weather was stored in the ground and returned to the rooms during mild weather. We also know that, other conditions remaining the same, heat flow to the ground increases with increases in moisture in the ground, but so far our information is qualitative not quantitative.

In closing I want to thank the A.S.H.V.E. for the cooperation given in this investigation. As Mr. Gordon mentioned, the investigation was sponsored by the *Institute of Boiler and Radiator Manufacturers* and the University of Illinois, but the sponsors agreed that the program should be of such a nature that it would supplement the investigations being made in the A.S.H.V.E. Laboratory. Members of the TAC on Panel Heating and Cooling have taken time to study our proposals before work was started and they have given valuable criticism of preliminary test reports that have been made to the TAC. We at the University, and I am sure the members of $I=B=R$, have appreciated this cooperation.


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No. 1499

EFFECTS OF NON-UNIFORMITY AND FURNISHINGS ON PANEL HEATING PERFORMANCE

By L. F. SCHUTRUM* AND C. M. HUMPHREYS**, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

SEVERAL YEARS ago a special room was constructed at the A.S.H.V.E. Research Laboratory for the study of heat transfer within a panel heated space. This room, commonly referred to as the Environment Laboratory, is approximately 12 ft wide and 24½ ft long, and the ceiling is adjustable to any height up to 12 ft. Either the floor or ceiling, or both, may be heated, and all six surfaces may be cooled to any temperature or combination of temperatures desired. Cooled air may be introduced at six locations in the room to simulate infiltration. A detailed description of the construction and instrumentation of this room is given in an earlier paper.[1]

In December 1952 a paper[2] was published giving the results of tests made in the Environment Laboratory with a heated ceiling, uniform environment (all surfaces other than the heated ceiling at a uniform temperature) and with and without infiltration air. In June 1953 a second paper[3] was presented, giving the results of a similar series of tests made with a heated floor. Some indication of the modifying effects of non-uniform environment and room furnishings on panel heating performance was given in one of these papers.[2] A more complete investigation of these effects has been made, and the results are presented herein.

EFFECTS OF NON-UNIFORM ENVIRONMENT

Ceiling Panel: Tests were made in the Environment Laboratory with several ceiling temperatures, various configurations of wall and floor surface temperatures, and with various infiltration rates and temperatures. The actual combinations of the above variables used in the tests are given in columns 3 through 8 and 10 and 11 of Table 1. In some of the tests one or more surfaces were neutral, that is, they were neither heated nor cooled, but were allowed to seek

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[1] Exponent numerals refer to References.

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TABLE 1—CEILING PANEL HEAT OUTPUTS AND ROOM AIR TEMPERATURES FOR NON-UNIFORM AND UNIFORM TEMPERATURE ENVIRONMENTS

(24½ × 12 × 8 ft high room)

Non-Uniform Conditions	Test No.	Room Surface Temperatures, F						AUST[a]	Infiltration		Panel Output Btu per (hr) (sq ft)		Room Air Temperature, F			
													Non-Uniform		Uniform[b]	
		Ceiling	Floor	North Wall	South Wall	East Wall	West Wall		Changes per hr	Temp F	Non-Unif.	Uniform[b]	30 in.	60 in.	30 in.	60 in.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
One long wall, cold.	187	89.8	64.9	64.1	64.7	45.6	64.7	60.5	none	—	28.3	29.0	61.7	63.3	61.6	62.8
	216	119.7	64.6	65.7	64.6	36.3	64.5	58.4	1.0	40.0	66.0	67.4	59.7	61.9	59.3	61.7
	217	119.8	64.7	65.7	64.6	36.2	64.4	58.4	none	—	68.0	67.0	60.5	63.5	60.7	63.3
	220	119.7	64.5	65.3	64.2	35.8	64.3	58.1	1.0	−0.6	68.0	68.7	56.6	59.7	55.7	59.7
	221	119.8	63.7	64.5	63.8	35.0	63.7	57.4	2.0	−2.4	69.7	70.1	51.4	55.6	50.0	55.6
One long wall, cold; Floor, neutral.	17	135.6	75.8*	69.4	67.7	28.0	67.2	61.8	none	—	87.7	84.2				
	18	105.9	65.9*	66.7	67.2	29.4	66.6	58.3	none	—	52.4	49.4				
	19	119.7	70.5*	67.2	66.0	29.1	66.0	59.5	none	—	67.8	65.4				
	20	119.8	75.2*	67.2	65.6	38.4	65.6	63.0	none	—	64.6	62.0				
	21	120.2	76.1*	67.1	65.4	46.5	65.4	65.0	none	—	62.4	61.2				
One long wall, cold; 3 walls, neutral.	26	129.1	63.0	74.2*	73.7*	27.8	73.4*	59.9	none	—	80.2	77.2	58.6	66.3	62.5	65.4
	27	110.9	63.3	69.1*	68.3*	27.2	68.0*	57.5	none	—	58.4	56.2	57.8	62.9	59.5	61.8
	28	142.1	65.1	79.0*	78.4*	33.0	78.5*	63.9	none	—	94.8	91.0	62.2	71.1	66.8	70.2
One long wall, neutral.	71	119.7	78.3	78.8	77.3	85.0*	77.4	79.5	1.0	−0.6	47.1	47.8	73.5	79.0	74.3	79.0
	72	119.8	77.8	79.1	78.0	83.5*	78.0	79.3	2.0	3.0	51.1	48.8	68.7	74.8	68.3	75.0
	73	120.0	77.6	79.4	77.6	89.4*	78.1	80.5	none	—	45.8	45.5	81.3	84.8	82.0	83.7
	197	120.5	65.2	65.8	64.9	79.1*	64.0	68.2	none	—	57.3	59.0	70.0	74.4	70.2	72.4
	198	119.6	65.2	67.6	65.2	77.2*	65.6	68.1	1.0	39.8	58.7	58.9	67.0	71.1	67.8	70.0
3 simulated windows, 34 Fc; 2 simulated outside walls; Floor, north & west walls, neutral.	114	109.7	79.6*	80.9*	66.3	66.7	79.6*	73.2	none	—	39.6	40.3	75.1	76.9	74.6	76.1
	115	89.5	71.8*	73.0*	67.9	67.9	72.0*	68.4	none	—	21.2	22.0	69.7	71.2	69.2	70.1
	116	106.7	75.2*	77.1*	67.5	67.6	75.6*	70.7	1.0	20.6	39.9	40.5	67.7	70.9	68.1	70.5
	227	90.3	71.7*	72.8*	68.0	67.2	71.9*	68.1	none	—	22.5	23.0	69.6	71.1	68.9	69.9
	228	90.0	68.5*	70.3*	66.9	67.0	69.5*	66.0	1.0	20.2	24.4	25.4	62.8	66.0	63.2	65.0

[a] Area weighted average temperature of the floor and walls.
[b] Uniform environment having the same ceiling temperature and AUST.
[c] One 4 × 5 ft window in center of south wall; two 4 × 5 ft windows in east wall.
* Neutral surface, neither heated nor cooled by liquid circulation.

their own temperature level. Neutral surface temperatures are indicated by asterisks.

The AUST, which is the area-weighted average unheated surface temperature, is given in column 9 for each test.

The average heat output of the ceiling was determined experimentally for each test made with *non-uniform* environment, and is listed in column 12 of Table 1. Listed for comparison in the next column is the heat output which would have been obtained for the same ceiling temperature, infiltration, and AUST, but with a *uniform* environment. These latter values were taken from figures in Reference 2. It may be seen that the values in columns 12 and 13 are in close agreement. The deviation is within ±5 percent for 33 of the 34 tests. The only exception is test No. 18.

The room air temperatures for non-uniform environment tests, taken at points 30 in. and 60 in. from the floor in the center of the room, are shown in columns 14 and 15 of Table 1. The next two columns indicate the air temperatures which would have prevailed for the same ceiling temperature, infiltration, and AUST, but with a uniform environment, as calculated from data given in an earlier paper.[2] Here again good agreement is shown between test results for non-uniform and predicted values for uniform environment. In 15 of the 18 tests recorded, the deviation in the air temperature at the 30 in. level is less than 1.5 deg. The deviation in temperatures at the 60 in. level exceeds 1.5 deg in only one test; however, in nearly all cases the deviation is positive, and indicates a temperature increase of approximately one degree attributable to non-uniform environment.

Floor Panel: A second series of tests was made to determine the effect of non-uniform environment with a heated floor panel. The test conditions, and the resulting panel heat outputs and room air temperatures are given in Table 2. Also given are comparable values of panel heat output and air temperature predicted for the same floor temperature, infiltration conditions, and AUST, but with uniform instead of non-uniform environment. These values for uniform environment were determined from data presented in another paper.[3]

A comparison of the panel heat output rates for non-uniform and uniform environment given in columns 12 and 13 of Table 2 indicates good agreement. In two of the 10 tests, the difference is slightly more than 5 percent. It is interesting to note that although the differences in heat outputs are small, the output for the non-uniform environment is greater in every case. A further analysis of these results seemed to indicate that for a given floor temperature and AUST, the panel output increased with an increase in non-uniformity in surrounding surface temperatures. However, the maximum deviation due to non-uniformity of surface temperatures was so small that this trend was not considered significant.

From a comparison of the values listed in the last four columns of Table 2, it is evident that the degree of non-uniformity of surrounding surfaces has little or no effect on the air temperature at the 30 in. and 60 in. levels in a floor panel heated space. The maximum deviation at the 60 in. level is only 0.5 deg.

EFFECTS OF ROOM FURNISHINGS

Two series of tests were made to determine the effect of room furnishings on the performance of floor panel and ceiling panel systems. In each series,

TABLE 2—FLOOR PANEL HEAT OUTPUTS AND ROOM AIR TEMPERATURES FOR NON-UNIFORM TEMPERATURE ENVIRONMENTS
(24⅓ × 12 × 8 ft high room)

NON-UNIFORM CONDITIONS	TEST NO.	ROOM SURFACE TEMPERATURES, F						AUST[a]	INFILTRATION		PANEL OUTPUT Btu per (hr) (sq ft)		ROOM AIR TEMPERATURE, F			
													Non-Uniform		Uniform[b]	
		Floor	Ceiling	North Wall	South Wall	East Wall	West Wall		Changes per hr	Temp F	Non-Unif.	Uniform[b]	30 in.	60 in.	30 in.	60 in.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
One long wall, cold.	164	84.6	65.2	65.9	64.8	45.3	65.0	60.8	none	—	34.3	32.8	68.4	68.4	68.8	68.4
	168	84.6	65.0	65.4	64.8	26.2	64.9	56.4	none	—	42.4	39.5	65.2	65.2	65.9	65.4
	173	84.9	71.0	69.9	69.8	36.2	69.8	62.8	1.0	0.8	38.4	36.6	64.8	66.7	66.8	66.7
	174	85.2	70.2	70.7	70.3	45.6	70.3	64.8	1.0	40.0	31.6	30.2	70.0	70.0	70.1	69.8
	175	84.7	70.3	70.1	69.9	45.6	69.9	64.7	2.0	39.6	32.6	31.2	66.9	67.9	68.2	68.2
One long wall, cold; 3 walls and ceiling, neutral.	214	85.0	74.8*	73.5*	73.3*	58.4	73.5*	70.6	none	—	20.5	20.2	75.3	75.2	75.5	75.2
3 simulated windows, 34 F[c];	223	85.6	73.4*	73.4*	67.5	67.3	72.0*	68.8	none	—	24.3	23.5	74.2	74.2	74.5	74.2
	224	80.2	70.6*	70.0*	67.0	67.4	69.3*	66.7	none	—	19.2	18.2	71.1	71.1	71.3	71.0
2 simulated outside walls;	225	80.4	70.7*	70.2*	67.5	67.7	69.3*	67.0	1.0	20.1	22.6	21.9	68.2	68.6	68.8	68.8
Ceiling, north and west walls, neutral	226	84.8	72.9*	72.2*	67.1	67.2	70.6*	67.9	1.0	20.1	28.5	27.6	70.6	71.3	70.8	70.8

[a] Area weighted average temperature of the ceiling and walls.
[b] Uniform environment having the same floor temperature and AUST.
[c] One 4 × 5 ft window in center of south wall; two 4 × 5 ft windows in east wall.
* Neutral surface, neither heated nor cooled by liquid circulation.

tests were made with different panel temperatures, AUST values, infiltration conditions, and degrees of uniformity of environment, and with two different amounts of room furnishings.

For most of the tests, the furniture consisted of a davenport, five office arm chairs with leather seats and backs, a large table, two small tables, and three lamps. This amount of furniture, which had a total projected area of approximately 200 sq ft, was judged to be representative for a living room of comparable size. In a few tests, four more chairs and four more small tables were crowded into the room, thus increasing the projected surface area of the furniture to about 330 sq ft. The projected area of a piece of furniture was taken as the sum of its area as seen from the top, the bottom, and each of the four sides.

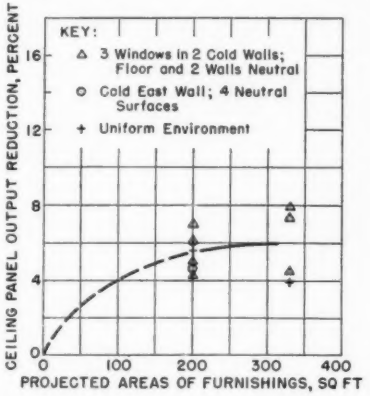


FIG. 1. EFFECT OF FURNISHINGS ON CEILING PANEL HEAT OUTPUT
(24½ × 12 × 8 ft high room)

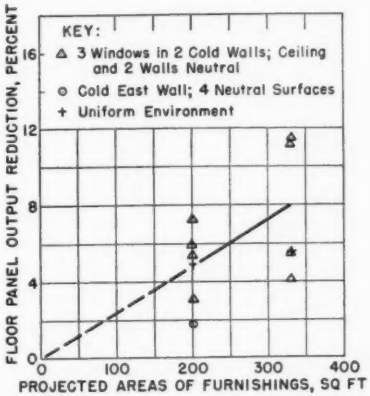


FIG. 2. EFFECT OF FURNISHINGS ON FLOOR PANEL HEAT OUTPUT
(24½ × 12 × 8 ft high room)

The introduction of furniture into the room caused a decrease in the heat output from both floor and ceiling panels. This decrease was directly proportional to the amount of furniture in the room, and was undoubtedly due to the obstruction provided by the furniture to direct radiant heat exchange between the room surfaces.

The amount of this reduction, expressed as a percentage of the panel output for an empty room having the same conditions of panel temperature, infiltration, and AUST with uniform environment, is shown in Figs. 1 and 2 for ceiling and floor panel heated rooms respectively. It is interesting to note that in both cases, the reduction in panel heat output due to a normal amount of furniture is in the order of 5 percent.

Although the points from which the curves of Figs. 1 and 2 were drawn show considerable scatter, the shapes of the curves are as might be expected. With a heated ceiling, the furnishings can screen the cool floor and the lower part of the cool wall but can have no direct effect on heat transfer to the cold surfaces above the furniture. Therefore, the total effect of the furniture rapidly

reaches a maximum as indicated by the reduction of slope in the curve with increasing area of furnishings. With a heated floor, the heat source is being screened from all of the cooled surface, and the curve shows no tendency to flatten within the limits of the tests made.

The surfaces of the furniture in a room are heated by radiation from the heating panel and heat is transferred by convection from these heated surfaces to the air, thus causing a higher air temperature in a furnished room than would be found in an empty room with comparable room surface temperatures. The increase in air temperature varies with the amount of furniture surface area in the space, and with the difference between the panel surface temperature and the AUST. The room air temperature increase caused by a normal amount of furniture in ceiling panel and floor panel heated rooms is shown in Fig. 3. The increase is greater with a ceiling panel than with a floor panel because convection already accounts for a considerable portion of the floor panel output.

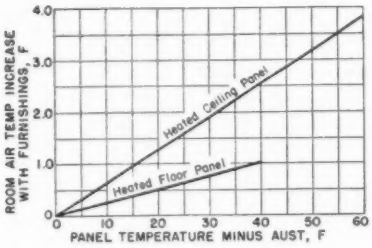


FIG. 3. EFFECT OF NORMAL AMOUNTS OF ROOM FURNISHINGS ON THE ROOM AIR TEMPERATURE AT THE 60 IN. LEVEL AT CENTER OF ROOM
(24½ × 12 × 8 ft high room)

The increases plotted in Fig. 3 were calculated from the dimensionless ratio (Air Temp—AUST)/(Panel Temp—AUST). See discussion of Figs. 4 and 5.

It should be noted that the decreased panel output and the increased room air temperature resulting from room furnishings were determined with the cold wall temperatures of the room held constant. This condition would not prevail in an actual installation. Instead, as the panel output decreased, wall surface temperatures, and therefore heat loss from the space and room air temperature would decrease somewhat. This would offset, at least to some extent, the trends noted above. The net effect of room furnishings on panel heat output and room air temperature may therefore be considered negligible for practical design purposes.

EFFECT OF FLOOR COVERINGS ON A HEATED FLOOR PANEL

To study the effect of floor coverings, experiments were conducted with three types of carpets: wool, rayon, and cotton. These carpets were selected by the manufacturer as being typical of the types and weights commonly used in residences. The carpets were used by themselves and in combination with two

types of carpet pads, one of hair and jute, and the other of rubber. The wool carpet was large enough to cover the entire floor of the room. Each of the others, rayon and cotton, covered one half of the room and were always used together to secure complete floor coverage. A general description of the carpets and pads is given in Table 3. The last column of the table gives the thermal conductance values for the carpets, pads, and the various combinations. Unless otherwise noted, these conductance values were determined from the experimental values of heat flow through and temperature drop across the floor covering. The values given are averages of the several tests made with each.

Table 4 lists the test conditions and the results obtained for a number of experiments with floor coverings. The experimentally determined average panel temperature under the carpet or pad is listed for each test in column 3.

The carpet surface temperatures given in column 7 were measured with a movable grid of thermocouples which was laid on the carpet surface. When

TABLE 3—DESCRIPTION OF FLOOR COVERINGS

CODE	DESCRIPTION	APPROX WEIGHTS OZ PER SQ YD	CONDUCTANCE BTU PER (HR) (SQ FT) F DEG
Carpet A	3 ply all wool	67	1.3
Carpet B	100 percent rayon, looped nap	65	0.8
Carpet C	100 percent cotton, straight nap	55	1.2
Pad A	40 oz, hair and jute	40	0.9[a]
Pad B	3/16 in. rubber pad	65	3.1[a]
Combinations:			
Carpet A and Pad A			0.53
Carpet A and Pad B			0.92
Carpet B and Pad A			0.42
Carpet B and Pad B			0.64[b]
Carpet C and Pad A			0.51
Carpet C and Pad B			0.87[b]

[a] These conductance values were determined from the results of tests made with various carpet and pad combinations.
[b] These values calculated from individual conductances.

in position the grid was slightly embedded in the carpet surface, but at least a part of the couples in the grid remained clearly visible to the observer.

The carpet surface temperatures given in column 8 were determined from the relationship previously established between AUST, temperature of a bare floor panel, and the room air temperature at the 60 in. level (see Figs. 7 and 10 of Reference 3). In a similar manner, the carpet surface temperatures in column 9 were determined from the established relationship between panel surface temperature, panel heat output, and AUST (see Figs. 1 and 4 of Reference 3).

The observed and calculated carpet surface temperatures in columns 7, 8, and 9 of Table 4 are in good agreement. This agreement demonstrates rather conclusively that the heat transfer from a carpet surface is essentially the same as the transfer from an ordinary bare panel surface at the same temperature.

The insulating effect of carpeting on a heated floor panel is evident from a comparison of panel and rug surface temperatures in columns 3 and 7 of Table 4. In test 262, Table 4, for example, the difference in these temperatures is

approximately 44 deg. This means that to get the same panel output with a wool carpet and a hair and jute pad, the temperature of the heating medium would have to be at least 44 deg higher than if the panel were bare.

It will be noted that only the wool carpet was used in the tests listed in Table 4. Results with the rayon and cotton carpet are not included because, as previously pointed out, each of these covered only half of the floor area of the room. However, the effect of these carpets can be readily predicted from

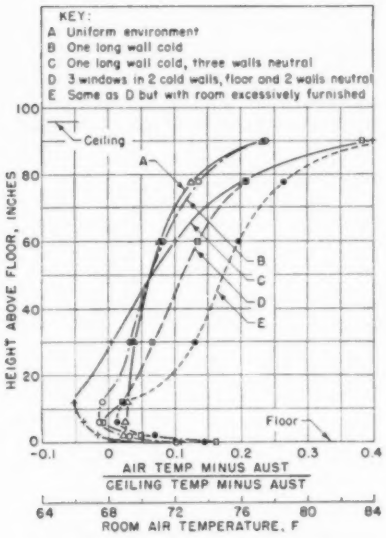


FIG. 4 *(above).* FLOOR TO CEILING ROOM AIR TEMPERATURE GRADIENT FOR A CEILING PANEL HEATED ROOM

(No infiltration. Values adjusted to AUST of 68 F and ceiling temperature of 108 F —24½ × 12 × 8 ft high room)

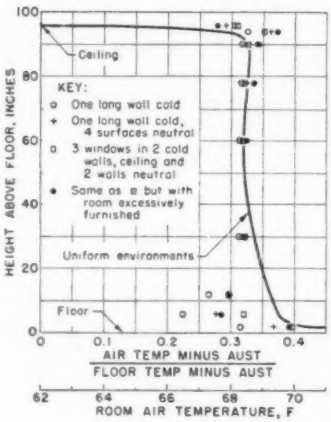


FIG. 5 *(above).* FLOOR TO CEILING ROOM AIR TEMPERATURE GRADIENT AT CENTER OF ROOM FOR A FLOOR PANEL HEATED ROOM

(No infiltration. Values adjusted to AUST of 62 F and floor temperatures of 82 F—24½ × 12 × 8 ft high room)

the conductance values given for them in Table 3. The temperature drop, ΔT, through any floor covering may be calculated from the equation,

$$\Delta T = Q/C$$

where

Q = heat flow through the floor covering, Btu per (hour) (square foot).
C = conductance of the floor covering, Btu per (hour) (square foot) (Fahrenheit degree).

Of the floor covering combinations listed in Table 3, the lowest conductance of 0.42 is for carpet B and pad A while the highest conductance of 0.92 is for

carpet A and pad B. The temperature drops required across these two combinations for a heat output rate of 25 Btu per (hr) (sq ft) would be as follows:

$$\text{Carpet B - Pad A} \quad 25 \div 0.42 = 60 \text{ deg}$$

$$\text{Carpet A - Pad B} \quad 25 \div 0.92 = 27 \text{ deg}$$

From this comparison, it is evident that care should be taken in selecting floor coverings to be laid on heated floor panels.

EFFECT OF VARIABLES ON TEMPERATURE GRADIENT

Fig. 4 shows the effect of non-uniform environment and furnishings on the floor-to-ceiling air temperature gradient in a ceiling panel heated room. All curves are adjusted to a 68 F AUST and a ceiling panel temperature of 108 F. The upper abscissa scale is a dimensionless ratio which may be used for any combination of ceiling panel temperature and AUST (also see Fig. 8, Reference 2 and Figs. 6 and 9, Reference 3). The lower scale relates the curves in terms of air temperatures.

Curve A for a uniform environment was taken from Fig. 8 of Reference 2. Curves B, C and D for varying degrees of non-uniformity in surface temperatures show some increase in gradient and some displacement to the right due to generally higher temperatures. Curve E, which is for the same conditions as curve D but with the addition of excessive furnishings, shows a distinct increase in air temperature.

Fig. 5 shows the effect of non-uniformity and furnishings on the floor-to-ceiling air temperature gradient in a floor panel heated room. The coordinates of the figure are similar to those of Fig. 4, but only the temperature gradient for uniform environment is plotted. The points for the other test conditions are shown. It will be noted that while there is considerable scattering of the points near the floor and some scattering near the ceiling, the temperature gradient from 30 in. above the floor to 6 in. below the ceiling is practically unaffected by non-uniformity of surface temperatures or by furnishings.

DISCUSSION OF RESULTS

Effects of Non-Uniformity and Furnishings: As previously indicated, the first testing programs in the Environment Laboratory were carried on under rather idealized conditions. All tests were performed in a given unfurnished room with all surfaces other than the panel at a uniform temperature. The results of this early work were of interest but could only be applied with reservation to actual design until a number of modifying factors could be studied. In reporting here on the effects of the non-uniformity of surface temperatures, room furnishings, and floor coverings, the work of the Research Laboratory is brought much nearer its ultimate goal, which is the development of simple and trustworthy design data on panel heating. Two other factors, the effect of room size and shape, and the effect of heating only a part of the floor or ceiling area, are still to be evaluated.

It has been shown that the degree of the non-uniformity of unheated surface temperatures makes no significant difference in the performance of a panel heating system, and that this performance can be predicted on the basis of the AUST. It has also been shown that the effect of room furnishings on panel

output and room air temperature may be neglected without appreciable error. These facts are indeed fortunate, for they will permit a considerable degree of simplification in design procedure.

TABLE 4—EFFECTS OF FLOOR COVERINGS ON HEATED FLOOR PANELS
(24½ × 12 × 8 ft high room)

TEST NO.	TEST CONDITIONS	PANEL SURFACE TEMP F	AUST F	FLOOR OUTPUT BTU PER (HR) (SQ FT)	ROOM AIR TEMP 60 IN. LEVEL	CARPET SURFACE TEMPERATURE, F		
						Measured[a]	Calc from Air Temp[b]	Calc from Heat Flow[c]
1	2	3	4	5	6	7	8	9
262	Uniform Environment No Infiltration Empty Room Carpet A, Pad A	124.5	64.7	23.3	69.8	80.6	80.6	81.7
263	Uniform Environment 1 Air Change at 20 F Empty Room Carpet A, Pad A	123.7	64.3	24.9	66.5	78.6	78.1	79.5
266	Non-Uniform Environment[d] 1 Air Change at 20 F Empty Room Carpet A, Pad A	127.3	69.7	24.5	71.2	82.1	82.3	84.5
267	Non-Uniform Environment[d] 1 Air Change at 20 F Furnished[e] Carpet A, Pad A	127.5	68.7	24.6	71.5	82.0	84.0	84.0
274	Uniform Environment No Infiltration Empty Room Carpet A, Pad B	100.2	64.8	19.9	69.5	78.0	79.5	79.3
273	Uniform Environment 1 Air Change at 23 F Empty Room Carpet A, Pad B	100.2	64.7	22.3	66.5	76.6	77.6	78.2
277	Uniform Environment No Infiltration Empty Room Carpet A	99.2	65.7	23.5	70.6	81.0	81.0	83.0
278	Uniform Environment 1 Air Change at 20 F Empty Room Carpet A	109.6	65.1	33.4	69.0	83.6	84.8	84.8

[a] Measured by thermocouple grid.
[b] Determined from Reference 3, Figs. 7 and 10.
[c] Determined from Reference 3, Figs. 1 and 4.
[d] 3 simulated windows in two outside walls, ceiling and two walls neutral.
[e] Normal amount of furnishings, 200 sq ft of projected area.

For purpose of illustration, let it be assumed that the following conditions of design have been determined:

AUST	68 F
Floor panel temperature	85 F
Infiltration	1 air change at zero F

From Fig. 1 of Reference 3, the heat output of the panel for an empty room with uniform surface temperatures and no infiltration equals 24 Btu per (hr) (sq ft).

From Fig. 4 of Reference 3, the increase in heat output due to infiltration as specified equals 6.7 Btu per (hr) (sq ft).

From this paper, corrections for room furnishings, and non-uniformity equals 0.0.

Therefore the total heat output which may be expected from the panel equals $24 + 6.7 + 0.0 = 30.7$ Btu per (hr) (sq ft).

This value may be subject to slight modifications when the results of current work on the effect of room size and shape, and the effect of heating only a part of the floor or ceiling area, become available.

Effects of Carpeting: The effects of carpeting on heat output from a floor panel are of interest, not only to the designer of floor panel systems, but also to the person living in a house heated by such a system. To the designer, the large difference in water temperature required for carpeted and uncarpeted panels emphasizes the importance of having a separate coil or circuit for each room with ample provisions for balancing or controlling each circuit. The ordinary adjustable cock frequently installed for balancing is not likely to prove satisfactory where such large temperature differences are required. At least partial compensation may be obtained by using closer tube spacing under areas which are to be carpeted.

The person living in a floor panel heated home should select floor coverings having high thermal conductances. For instance, a rubber pad has much less effect on the operation of the heating system than a hair and jute pad.

The results given in this paper are for a floor completely carpeted. The effects of carpeting would decrease as the percentage of the carpeted area of the panel decreases. In rooms where most of the panel area is carpeted, it is quite likely that the temperature of the uncarpeted areas would be uncomfortably high.

With other than floor panel type heating systems, carpeting serves only as an insulation applied to the floor surface. This is particularly effective where floors are laid directly on the ground or are otherwise exposed.

CONCLUSIONS

All of the tests from which the following conclusions are drawn were made in the Environment Laboratory. However, these tests were made under such a variety of conditions that it seems reasonable to assume that they may be applied with satisfactory accuracy to any ordinary structure.

1. The performance of a panel heating system in a space having a non-uniform surface temperature environment can be predicted with satisfactory accuracy on the basis of the area weighted average unheated surface temperature (AUST) of the space.

2. Furnishings in a panel heated space tend to reduce the heat output of the panel, and increase the room air temperature. However, the net effect of these trends may be considered negligible for design purposes.

3. When a floor covering is laid over a heated floor panel, the surface temperature of that panel and the temperature of the heating medium must be considerably increased to maintain the same heat output to the space that would be obtained from the bare panel. For a heat output of 25 Btu per (hr) (sq ft), the amount of this temperature increase was found to vary from 27 deg to 60 deg for the various combinations of carpets and pads tested.

4. Coverings for heated floor panels should be selected to provide a minimum of resistance to heat flow from the panel to the space.

ACKNOWLEDGMENT

Acknowledgment is made to the Bigelow-Sanford Carpet Co., Inc., which supplied all of the carpets and carpet pads used in these experiments.

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1. The A.S.H.V.E. Environment Laboratory, by Cyril Tasker, C. M. Humphreys, G. V. Parmelee, and L. F. Schutrum (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, Vol. 24, March 1952, p. 119).

2. A.S.H.V.E. RESEARCH REPORT No. 1473—Heat Exchanges in a Ceiling Panel Heated Room, by L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 197).

3. A.S.H.V.E. RESEARCH REPORT No. 1490—Heat Exchanges in a Floor Panel Heated Room, by L. F. Schutrum, G. V. Parmelee, and C. M. Humphreys (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 495).

DISCUSSION

W. P. CHAPMAN, Pittsburgh, Pa.: The figures that were given showing the decrease in panel output for various amounts of furnishings, and also an increase in air temperature, would, at first glance, appear that if you put enough furniture in there you wouldn't have any output but would have a very hot room. That, of course, is impossible. However, the *required* panel output is decreased by furniture and the *required* air temperature is increased by furniture. In other words, to get the same air temperature, the *required* output is reduced. It is easier for me to use this terminology *required* than the way it was first presented to me a few months ago, and as it appears in this paper.

The question that I have is concerned with the non-uniformity that was suggested by Sartain and Harris in their paper;* *i.e.*, that the effect of windows and possibly the draft coming off the windows and the path of the infiltration increased the output towards the outside wall. It showed quite a large increase in the Harris and Sartain paper and one of the commenters on the paper suggested that it was due to a higher water temperature over those pipes. I wonder if a similar report could be given on this paper. In other words, under the long cold wall did the meters closer to the cold wall show an increased output as compared to the average, and was it due largely to higher convection?

R. J. MINDAK, Chicago, Ill.: If you analyze the problem of increased output near the edge using simple radiation calculation methods, it can be shown that the edge must have increased output. By evaluating the geometrical configuration factor of the

* Heat Flow Characteristics of Hot Water Floor Panels, by E. L. Sartain and W. S. Harris (A.S.H.V.E. TRANSACTIONS, Vol. 60, 1954, p. 103).

ceiling near the wall and the configuration factor of the center of the ceiling with respect to the wall you can see that the edge of the ceiling will have a higher output. Thus simple theory shows that we might expect these results.

These furnishings all have an average height and I am wondering, if you decrease the height of the room, so that it would be equivalent to the height of the room that you would have, measuring it from the ceiling to the average height of the furnishings, whether you would have this same effect on the output.

In tests we have run at Armour on non-uniform environment with one long cold wall, our results are in very good agreement with the uniform output measurements presented in the previous paper last year.

P. R. ACHENBACH, Washington, D. C.: It seems to me that the data in Table 1 show that the non-uniform environment has little effect on the panel output as compared to the uniform environment. It indicates the relative unimportance of convection on the heat transfer from a ceiling panel. I recall when the paper on heat transfer with a uniform environment was presented a year ago a question was raised about the lack of convection currents in the room. This paper now indicates that the convection currents that are produced by one or more cold walls have little effect on the heat transmission from the warm ceiling surface.

The data in this paper show that non-uniformity of temperature of the heat absorbing surfaces in the Environment Laboratory affected the heat output of the heated panel no more than 5 percent in most cases. In the test room there was almost exactly twice as much total wall area as ceiling or floor area, yet the weighted-area method of determining the AUST gives equal weight to a square foot of each type of surface. The shape factors for the test room are such that more heat is transferred by radiation per square foot of area between ceiling and floor than between ceiling and walls or between floor and walls. Is it not possible that for rooms of other sizes or shapes that the disparity in panel output between the uniform and non-uniform environment conditions would be greater than was observed in the test room?

JOHN EVERETTS, JR., Philadelphia, Pa.: The results of this paper indicate a loss of approximately seven percent in output and also a very good insulating effect for carpet. I feel it is obvious that you are going to have greater heat loss to grade or a crawl space with carpeting than without. Do the authors have any data on this?

E. L. SARTAIN, Urbana, Ill.: In response to Mr. Everett's question concerning the increase in reverse losses for carpeted panels, information has been obtained from tests conducted at the Floor Slab Laboratory, University of Illinois. In several of the test rooms in the Laboratory the concrete floor panels were covered with various combinations of carpeting and pads.

It was found that in one room where a combination of heavy weight carpet and heavy weight pad was used, there was an increase in fuel consumption of about 40 percent at design conditions of 70 F indoor and −10 F outdoor temperatures. However, at lower values of indoor-outdoor temperature differences there was evidence of reverse flow from the panel and less fuel consumption than would be expected with an uncovered floor panel. When the data were integrated over the entire range of indoor-outdoor temperature differentials experienced in Urbana, Ill., during the heating season, it was found that the annual fuel consumption was only about 5 therms, or 4 percent, greater for the carpeted floor panel than for the bare floor panel.

AUTHORS' CLOSURE (C. M. HUMPHREYS): Mr. Chapman asked if greater heat exchange per unit area was observed at the edges of the heated floor along the cold walls than near the center of the floor. The increased heat exchange along the cold walls was so great that additional heat flow meters were placed along the edges of the floor to properly indicate this effect. As suggested by him, this increased output was due largely to the increased convection at the bottom of the cold walls.

Some increase in output along the edges would also be expected because of the increased radiation to the adjacent cold wall surfaces.

We wish to thank Mr. Sartain for answering the question asked by Mr. Everetts regarding the effect of carpeting on the heat loss to the ground. The work at the Research Laboratory did not provide an answer to this question.

Mr. Achenbach asked if the effect of non-uniform environment might be greater in rooms of other sizes or shapes than in the one tested. The modifying effect of room size and shape is now under study, and a definite answer to this question should be available in the near future.

A word of explanation might be in order regarding the values for floor panel heat outputs for uniform environment shown in Column 13 of Table 2. Some of you may have noted that these do not check exactly with the values given in an earlier paper.† Additional work in the room has resulted in further refinements in correction factors, and values from earlier tests have been recalculated for the comparisons presented in Table 2.

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† See Reference 3 of this paper.



1500

No. 1500

FIELD STUDIES OF FLOOR PANEL CONTROL SYSTEMS‡—PART II

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MINNEAPOLIS, MINN.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Minnesota, Minneapolis, Minn.

THE FIELD STUDIES on floor panel control systems have been in progress the past four years as part of the A.S.H.V.E. research program on panel heating. This phase of the program has had the advisory guidance of *Group D* on Controls, of the Technical Advisory Committee on Panel Heating and Cooling.†† The purpose of the studies was to determine from field investigation, the various factors affecting control stability and the ability of various control systems to overcome or assist these factors. The installations tested were in various types of building structures having varying percentages of glass area.

The criterion selected for the field studies was stability of controlled temperatures, since the effectiveness of a control system is primarily based on the conditions maintained. If control stability is attainable, it is then possible that these conditions necessary for comfort will likewise be attainable.

The results of two years of field studies, together with complete construction and instrumentation details were presented in a previous paper.‡

The stability of a control point is not to be interpreted as meaning a constant temperature, but rather a temperature which may vary in some relation with load conditions. The principal factors constituting the load conditions are: (1) outdoor dry bulb temperature; (2) solar radiation; (3) wind; (4) building construction (glass area, infiltration, thermal transmission and heat storage capacity); and (5) occupancy.

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‡ Field Studies of Floor Panel Control Systems, by A. B. Algren, E. F. Snyder, Jr., and J. S. Locke (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 173).

Presented at the 60th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Houston, Tex., January 1954.

To facilitate the field studies so that basic information could be obtained for early dissemination to the profession, the following three types of construction only were selected:

Type A Installation—Heavy floor panel, heavy building construction, large glass area. This *Type A* is a single-story general office in a factory building located in Milwaukee. Instrumentation of this installation was unchanged for the 1952-53 heating season.

Type B Installation—Heavy floor panel, light building construction, large glass area. *Type B* is the factory section for a second Milwaukee plant; the revised piping layout for this installation is shown in Fig. 1. The control systems furnished for the 1952-53 heating season were the same as for the 1951-52 heating season except for

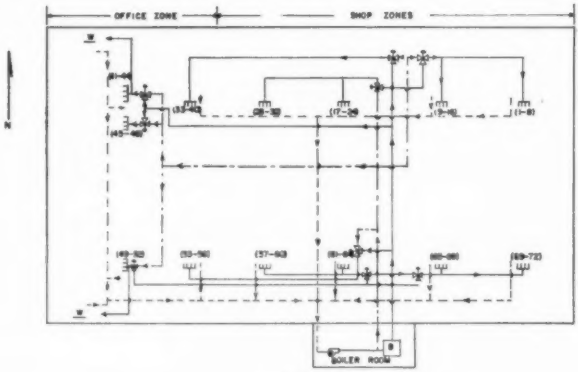


FIG. 1. PANEL LAYOUT FOR *Installation B*

the variable locations of the outdoor temperature sensing bulb; in addition, a second boiler was added during the summer of 1952 to increase the plant heating capacity.

Type C Installation—Heavy floor panel, light building construction, small glass area. *Type C* is a residence located in Minneapolis. The control system of this installation for the 1951-52 heating season was on-off valve control and was changed to a proportioning three-way mixing valve control for the 1952-53 heating season. Fig. 2 gives a wiring diagram of the control system.

TESTING PROCEDURE

The test program for the 1952-53 heating season was specifically planned to consummate the floor panel studies. Certain changes in instrumentation were made for one or more of the following reasons: (1) to correct or improve design features of the heating system, (2) to increase the facilities for additional studies, and (3) to isolate control variables.

As no instrumentation changes were made at *Installation A*, the studies at this plant were essentially those of supplementing and verifying previous test results.

Individual test periods were reduced to four days for expedience of analysis, so that a maximum amount of necessary rechecking could be accomplished. The temperature trends from the desired portion of each strip chart were recorded in chart form.

IMPORTANT FACTORS FOR CONTROL OF SYSTEMS

Various factors of primary importance in the control of floor panel heating systems were studied throughout the field test program. These factors, which

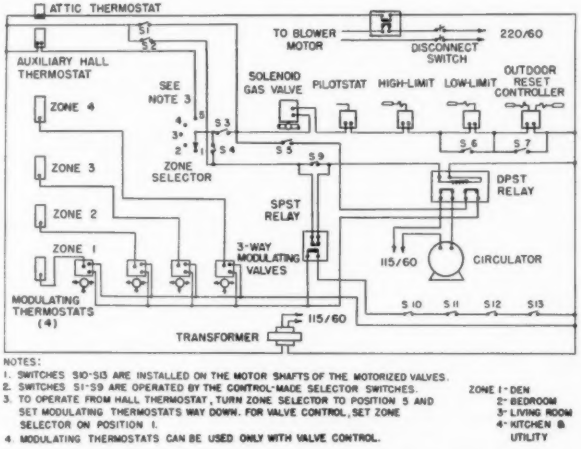


FIG. 2. SCHEMATIC HEATING CONTROL CIRCUIT FOR *Installation C*

are either peculiar to, or their effect more pronounced in floor panel heating systems, are listed in the order of discussion:

1. Rapid changes of heating load.
2. Thermal inertia of floor panel slab.
3. Use of solar compensator.
4. Location of outdoor temperature sensing bulb.
5. Common return water headers.
6. Thermal inertia of solar compensator.
7. Boiler water temperature control: (*a*) fixed; (*b*) reset.
8. Constant *vs.* variable heating load.
9. Exposed walls, effects on: (*a*) control; (*b*) temperature distribution; (*c*) air motion.
10. Floor covering.

Due to the inherent mass of the floor panel slab, the control problem associated with sudden heating load changes is of far greater consequence in a floor panel heating system than in any other type. The large thermal inertia

of the floor panel slab precludes a rapid change in the energy level of the slab. Some outside weather conditions which effect a rapid change in heating load are sudden changes in: (1) the intensity of solar radiation; (2) the outside temperature; and (3) wind velocity, and/or direction.

The effect of a change in intensity of solar radiation on the space temperature is more pronounced in a construction where the walls that are exposed to the sun have large glass areas.

USE OF SOLAR COMPENSATOR

A solar compensator, which had been previously developed for another use, was employed in conjunction with the control system which utilized reset boiler water temperatures. This control system permits adjustment of the boiler water temperature in accordance with the sensed outside temperature. To account for solar effects, inclusive with heating load, the outdoor temperature sensing bulb was located in the solar compensator.

The solar compensator consists of an insulated enclosure with glass in one side mounted in the same plane as the glass in the building. It is designed to take into account the immediate solar gain through the glass area of the walls and is oriented in the same direction as the heating zone it serves. Solar energy passing through the compensator glass impinges on a black plate in the compensator enclosure. This prevents outward reflection of the solar waves and results in an increase in the interior compensator temperature above the outdoor ambient temperature in proportion to the solar energy received. The outdoor temperature sensing bulb is located within the solar compensator back of the black plate and thus responds to the temperature in the solar compensator. The window area of the compensator is fitted with a shade so that its effective glass area can be adjusted in accordance with the amount of glass area in the building.

When the outside temperature sensing bulb is located in the solar compensator and there is an increase in solar intensity, the heating load decreases and the boiler water temperature would be adjusted downward. With either proportioning or on-off valve operation the thermal energy supplied to the panel would be decreased. However, because of the thermal inertia of the panel slab the rate of heat supplied to the space is greater than the heat loss. The space temperature rises until equilibrium is reached. This same inertia effect exists in reverse when a rapid increase occurs in the heating load due to a decrease in solar intensity. Both conditions are shown for *Installation B*, in Fig. 3, for the period from 8:00 a.m. to 4:00 p.m., Feb. 4, 1953. With the sudden increase in solar intensity, the energy to the panel was reduced sharply, as may be seen by observing the boiler and panel water temperature, although, the space temperature increased while the reverse condition occurred when the solar intensity decreased.

The space temperature varies more under the same load conditions if the outdoor temperature sensing bulb is not located in solar compensator but is shielded from the sun so as to sense only the dry bulb temperature. In this case the boiler water temperature is adjusted only in accordance with the dry bulb and does not sense the solar effect which immediately reduces the heating load.

Fig. 4 shows test results for the same installation, and for the same control system as shown in Fig. 3, but with the outdoor temperature sensing bulb in

a different location. In Fig. 4, the period between 9:00 a.m. and 6:00 p.m., January 11, 1953, is comparable to the previously mentioned period of Fig. 3 in heating load, though the increase in heating load is not as sharp. The boiler water temperature was not adjusted downward to the same extent as previously discussed; consequently, the space temperature showed a greater increase and remained above the space thermostat setting for a longer period of time.

In addition, in Fig. 4, there is a period from 5:00 a.m. to 3:00 p.m. January 12, 1953, in which the outside temperature increased markedly, and no sunshine occurred. With the reduced heat supply to the panel, the thermal inertia of the slab increased the space temperature from 71 F to 75 F. The sudden decrease in space temperature from 75 F to 72 F is due to the fact that the clerestory windows were open and the 24 in. diameter exhaust fan was turned on.

LOCATION OF OUTDOOR TEMPERATURE SENSING BULB

Excluding the uses of a solar compensator, a problem common to all systems of control is that of location of the outdoor temperature sensing bulb. The considerations involved in determining the location for optimum effects are: (1) type of construction involved in the installation; (2) location on sunny side or shaded side of installations, shielded or unshielded; and (3) location and ability to sense prevailing winds.

The location is particularly important for any installation which includes large glass area exposed to the sun, as in *Installation B*.

Location for the 1951-52 Heating Season: Through the 1951-52 heating season, the outside temperature sensing bulb was located on the west side of the boiler room, and was thus unable to sense the early solar effects. Therefore, energy was supplied to the panel longer than was necessary. The result of this was the overheating of the interior space with complaints from the occupants.

An example of this condition of overheating due to lack of proper sensing of the solar effect is shown in Fig. 5, which represents the period from 10:00 a.m. January 29, to 10:00 a.m. February 1, 1952. Observe especially the period from 8:00 a.m. to 3:00 p.m. January 31, 1952, in which the increasing outside temperature together with the solar effect caused a considerable part of the space temperature increase from 70 F to 80 F.

COMMON RETURN WATER HEADERS

A control problem in general will result from connecting the water returns of the separate zones to a common return header. It is seen that for period referred to in Fig. 5 a temperature drop occurs through the panel even though only return water is being circulated. This is due to the fact that a common return header was used for all zone heating coils, and the return water from a zone calling for heat was at a higher temperature than the water temperature requirements of the zone represented in Fig. 5. This accounts in large part for the marked overheating.

During the 1952-1953 heating season for the *Installation B* two separate control systems were employed, and for each control system, the outdoor temperature sensing bulb was placed in several locations for the various testing periods. The predetermined schedule of changing control systems and the lack

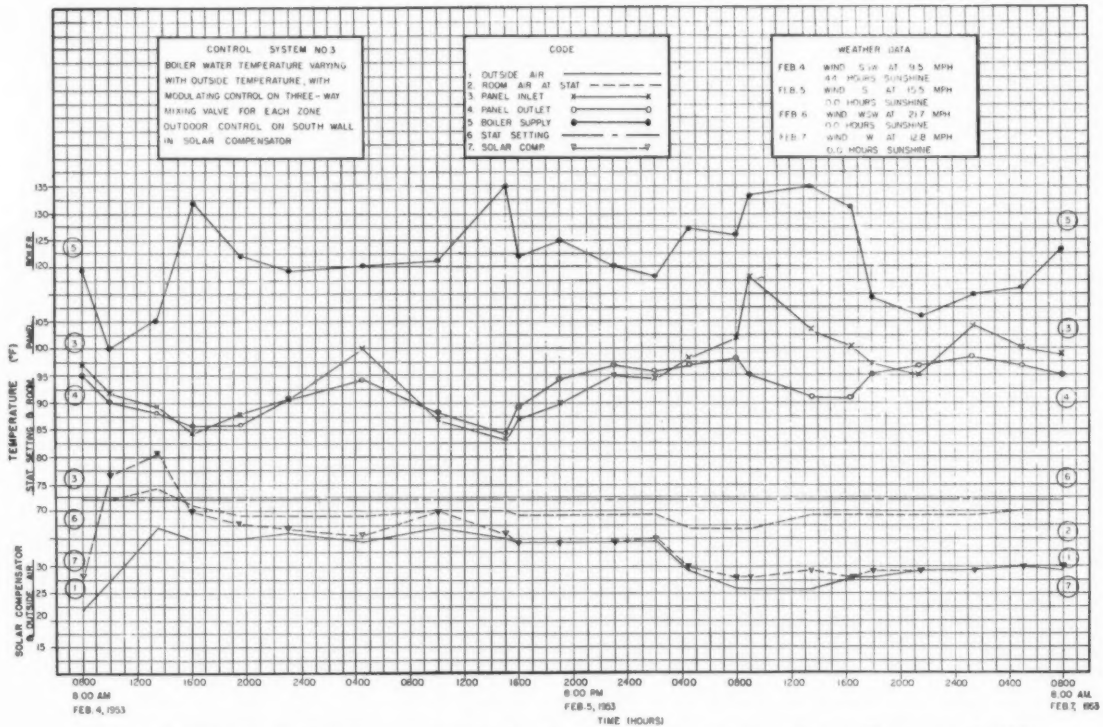


FIG. 3. TEMPERATURE TRENDS FOR *Installation B*, 8 A.M. FEBRUARY 4 TO 8 A.M. FEBRUARY 7, 1953

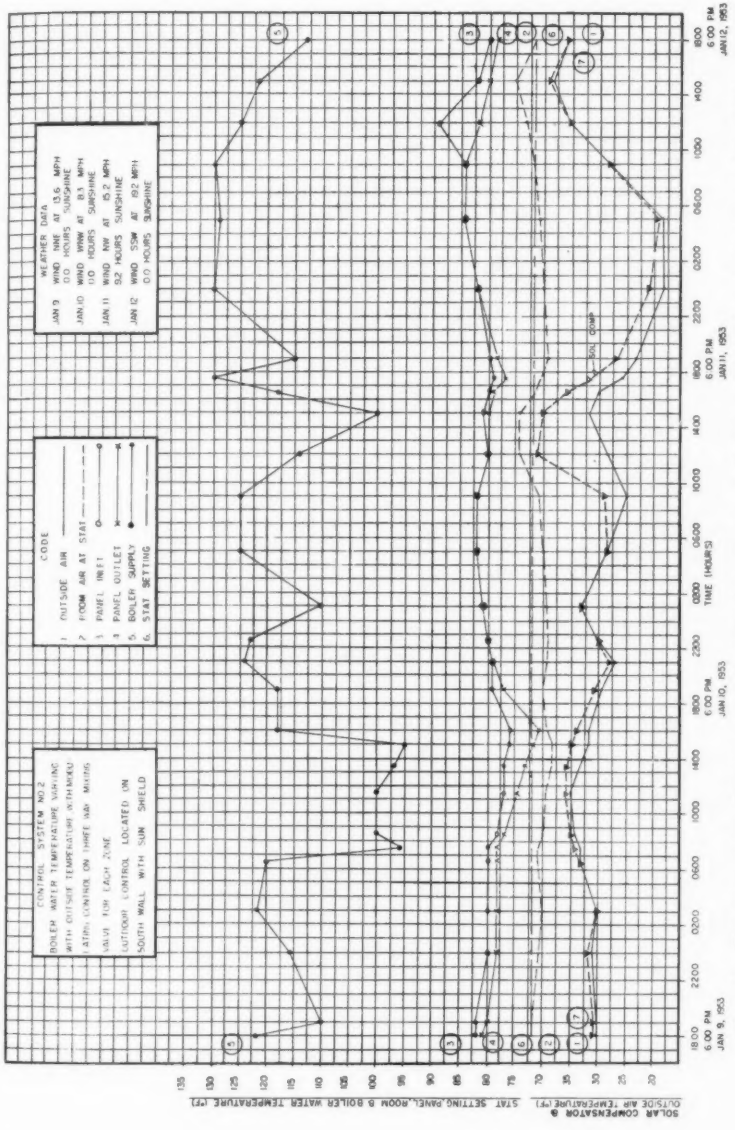


FIG. 4. TEMPERATURE TRENDS FOR *Installation B*, 6 P.M. JANUARY 9 TO 6 P.M. JANUARY 12, 1953

of proximity of the engineer supervising the control changes did not permit continuous scrutiny of the weather.

In subsequent evaluation of the strip charts, it was found that solar conditions were not identical and did not provide a complete comparison of the two control systems. However, the advantage of locating the outdoor temperature sensing bulb to detect the solar effect can be seen by comparing the period of Fig. 5, in which the outdoor temperature sensing bulb was located on the west wall, with that period of Fig. 4, discussed in the foregoing, in which the outdoor temperature sensing bulb was located on the south wall of *Installation B*.

THERMAL INERTIA OF SOLAR COMPENSATOR

When there was no sunshine, the temperature in the solar compensator should have been the same as the outside temperature. Solar compensator temperature in Figs. 3, 4 and 6 show that this was not always the case. Apparently the thermal inertia of the compensator accounts in great part for the difference when the solar intensity is decreasing. Since the glass area of the solar compensator is exposed directly to the sun, no thermal lag is evident for an increasing solar intensity. A review of the charts will show that a difference occurs immediately after the solar effects ceases.

Fig. 6 is more representative of the response of the boiler water temperature to the solar effect when the outside temperature sensing device is located in the solar compensator. A drastic drop in the boiler water temperature and the panel temperatures as the solar intensity increases can be seen in Fig. 6 by noting the periods from 8:00 a.m. to 4:00 p.m., February 8, and 8:00 a.m. to 12:00 noon, February 9. The space temperature as a result, experienced a much smaller increase than if the outside temperature sensing bulb had been located outside the solar compensator and shielded from the sun.

BOILER WATER TEMPERATURE CONTROL

Fig. 7 represents a three-day period of operation with the same control system shown in Fig. 6, except that the boiler water temperature was held constant. The downward adjustment of the boiler water temperature and the panel temperature with an increasing outside temperature is clearly seen in Fig. 6, for the periods previously discussed. A similar outside temperature increase for the period between 8:00 a.m. and 2:30 p.m. December 18 is shown in Fig. 7. It is observed that prior to this period a steadily increasing load demand was imposed on the boiler and that the temperature of the water in passing through the boiler never reached the boiler setting temperature. As the outside temperature started upward, the load demand on the boiler was decreased and the boiler water temperature quickly reached the boiler temperature setting. Likewise, it is seen that the panel temperature increased. The space temperature increased 5 deg two hours before the peak outside temperature occurred, and was prevented from additional increase by the opening of the clerestory windows and the starting of the exhaust fan.

The extent to which a given control system can maintain a set temperature in a given enclosed space depends on many factors. Normally, the largest single factor is that of the type of weather load condition against which the control system is operating. For purposes of discussion, load conditions will be

roughly classified according to outside dry bulb temperatures as follows: (1) mild—above +40 F; (2) medium—+10 F to +40 F; (3) severe below +10 F.

CONSTANT *vs.* VARIABLE HEATING LOAD

For any constant load condition, provided the heating plant is of sufficient capacity, the space temperature can be maintained fairly constant at the set temperature. This is almost entirely dependent on the sensitivity of the thermostat. The ability of a control system to maintain a constant space temperature for a constant load condition is seen in Fig. 8. For the first two days of this period, the outside temperature was essentially constant at about 33 F, while the space temperature remained essentially at 72 F.

Fig. 8 can be compared with Fig. 9, for which the same control system was employed for a variable outside load temperature, in which the minimum was 6 F and the maximum was 40 F. Figs. 4 and 7 (*Installation B*) also represent a comparison for relatively constant and variable outside load conditions as do Figs. 10 and 11 (Zone 1 of Minneapolis installation).

From these three comparisons it can be seen that the space temperature is most erratic when the outside temperature is varying rapidly.

The term MRT is defined for this paper as the mean radiant temperature of the inside surfaces of the building, and more specifically is the weighted average temperature of the heated and unheated surfaces. The MRT felt by the individual would depend on his position with respect to the inside surfaces and would increase as the individual moved inward from the exposed walls, as the floor surface would be essentially constant throughout the building. The direct radiant effect of panel heating in regard to the space temperature increases with an increase in the ratio volume/exposed wall area, or the less the exposed wall area, the less difficult becomes the problem of maintaining a set space temperature.

EFFECTS OF EXPOSED WALLS

In view of the fact that the exposed walls and windows have a direct influence on the temperature distribution and that the room thermostat is confined to sensing temperature at only its location, studies were made at each of the three installations to determine air temprature distribution in the space and air flow patterns around the exposed windows. Air velocities along the exposed windows were obtained with a hot wire anemometer. The profile of these velocities was obtained by smoke observation.

Air Temperature Distribution: Air temperature distribution was obtained by fixed and shielded thermocouples located on a portable stand.

Figs. 12 and 13 show vertical temperature distribution at various positions in the General Office and Engineering Office, respectively, of *Installation A*. It is noted that at each position the air temperature is higher 3 in. above the floor and 3 in. below the ceiling (117 in. level) than at intermediate levels. This is due to the convective effect on the air by the warm surface of the floor and ceiling. The intervening space between the floor and ceiling, which is not affected by localized convection, shows a very even vertical temperature distribution.

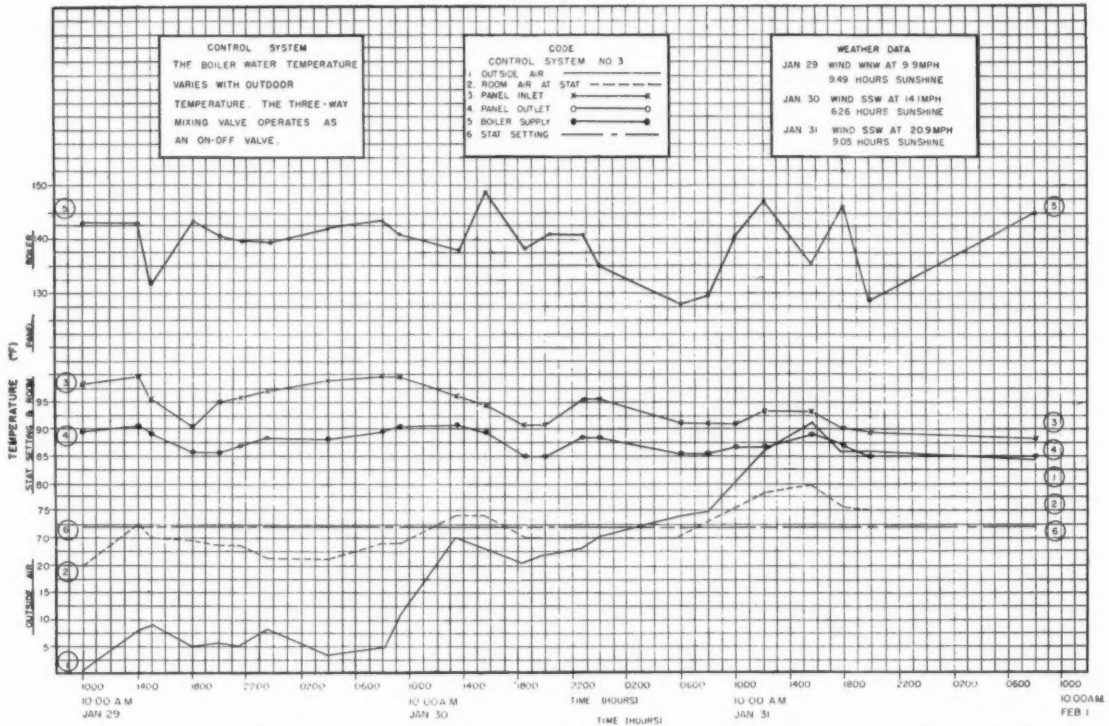


FIG. 5. TEMPERATURE TRENDS FOR *Installation B*, 10 A.M. JANUARY 29 TO 8 A.M. FEBRUARY 1, 1952

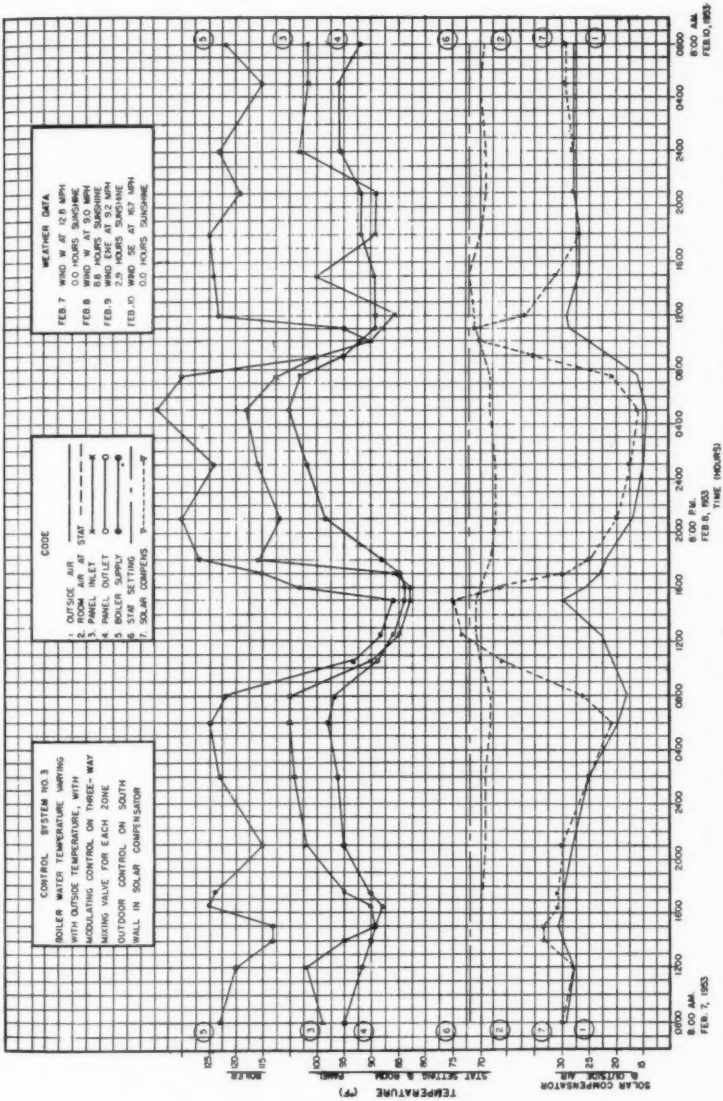


FIG. 6. TEMPERATURE TRENDS FOR *Installation B*, 8 A.M. FEBRUARY 7 TO 8 A.M. FEBRUARY 10, 1952

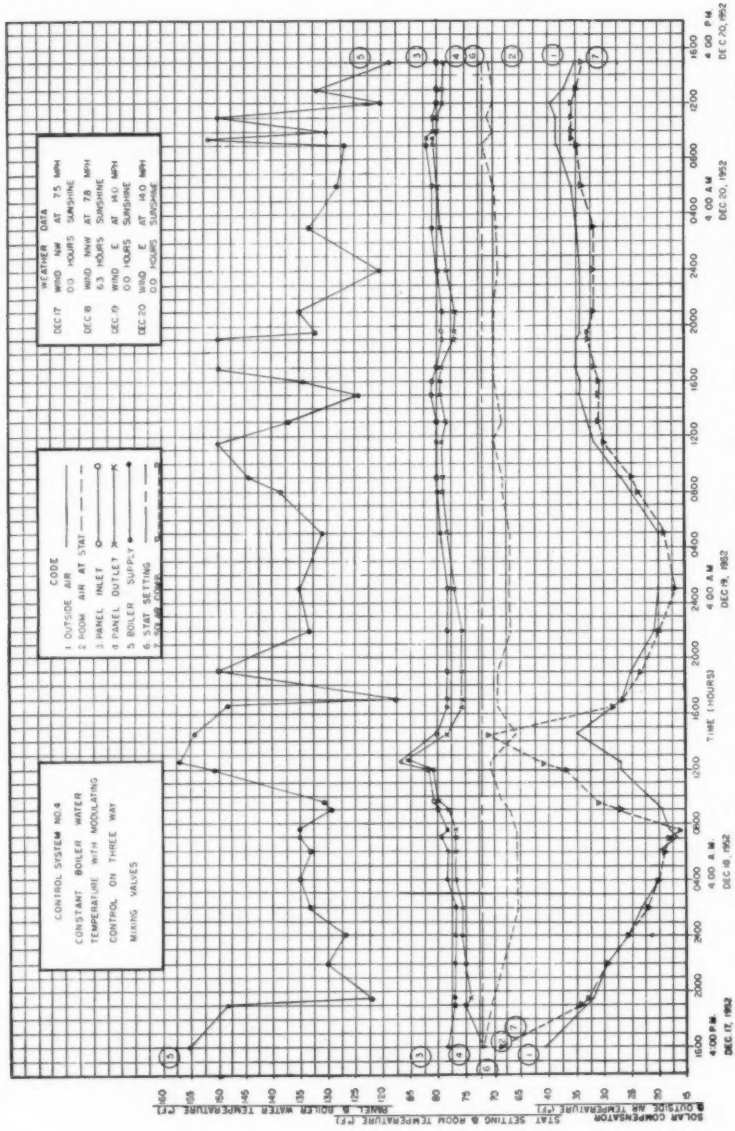


FIG. 7. TEMPERATURE TRENDS FOR *Installation B*, 4 P.M. DECEMBER 17 TO 3 P.M. DECEMBER 20, 1952

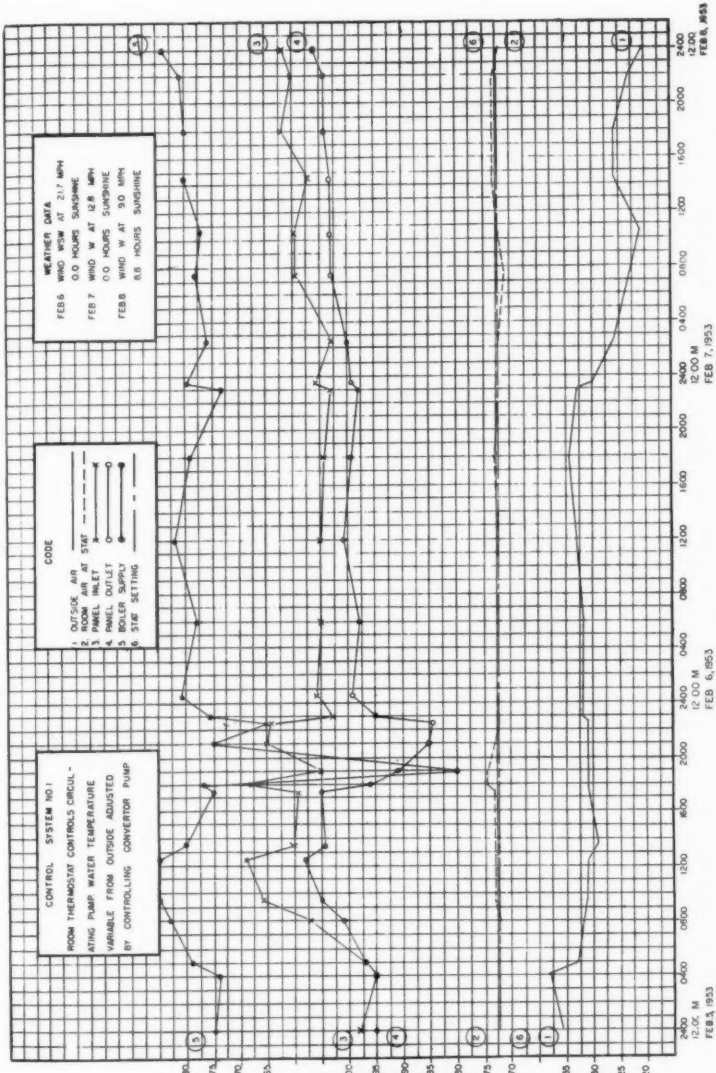


FIG. 8. TEMPERATURE TRENDS FOR *Installation A*, FEBRUARY 5 THROUGH FEBRUARY 7, 1953

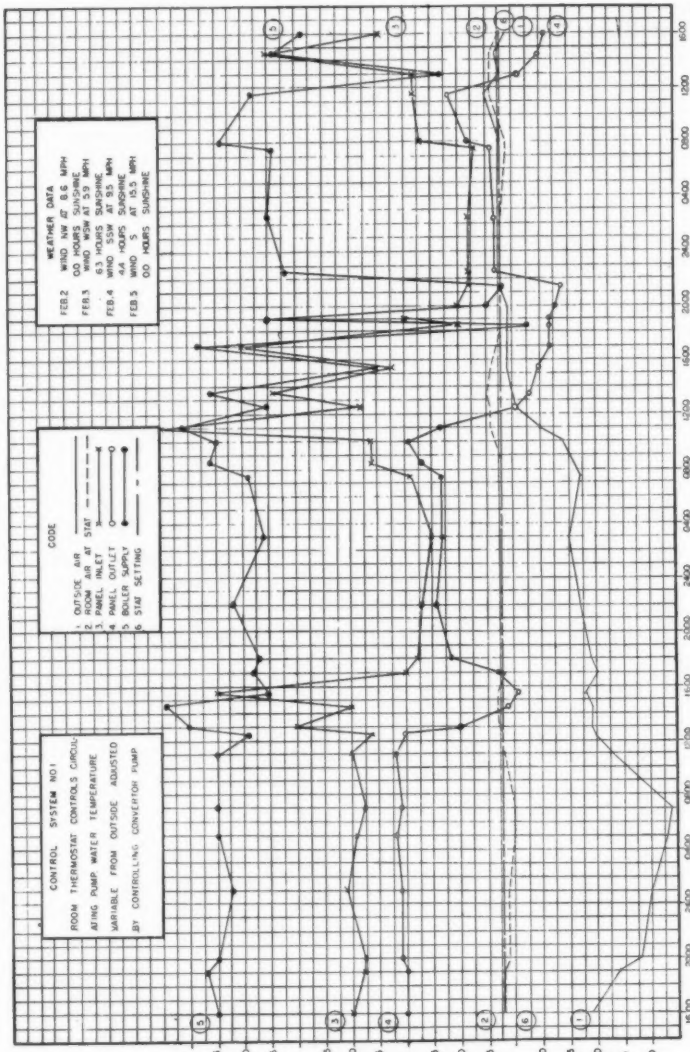


FIG. 9. TEMPERATURE TRENDS FOR *Installation A*, 4 P.M. FEBRUARY 2 TO 4 P.M. FEBRUARY 5, 1953

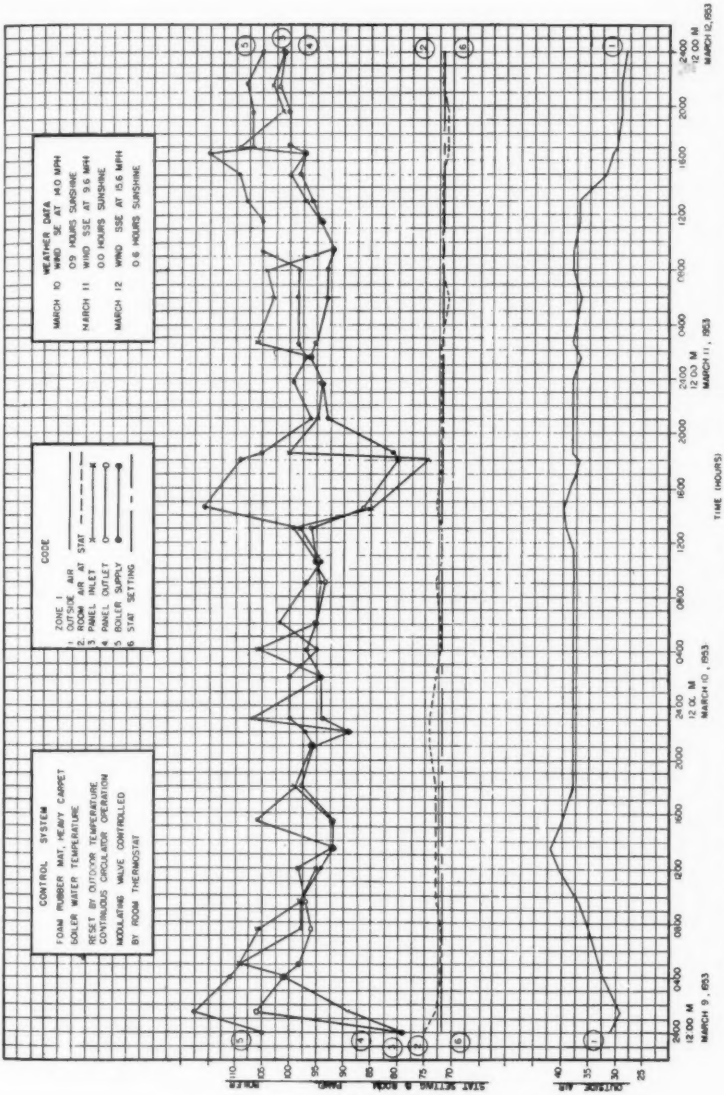


FIG. 10. TEMPERATURE TRENDS FOR *Installation C*—ZONE 1, MARCH 9 THROUGH MARCH 11, 1953

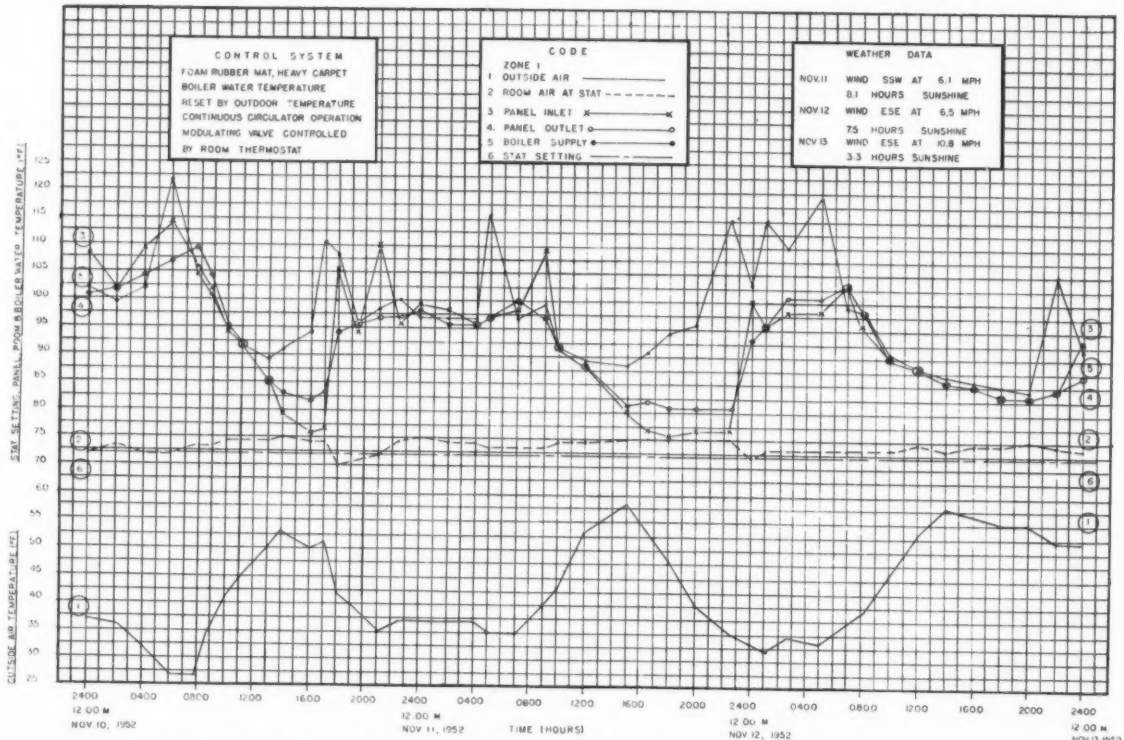


FIG. 11. TEMPERATURE TRENDS FOR *Installation C*—ZONE 1, NOVEMBER 10 THROUGH NOVEMBER 12, 1952

The short distance between the exposed window and the inside partition precluded the study of *Installation A* of the offsetting effect of the heated floor on the down draft as a function of distance from the window. However, this factor was studied at *Installation B* by taking air temperatures at various distances from the window at specific levels. Fig. 14 shows the effectiveness of the heated floor in heating the air moving downward and inward in a plane perpendicular to the window. The temperature distribution vertically may be observed at specific levels as indicated by Fig. 14.

Air Motion: The rate of air temperature change at the various levels tested is greatest at the 3 in. level and becomes progressively less at the 30 in., 96 in., and 60 in. levels as the distance from the outside wall is increased up to

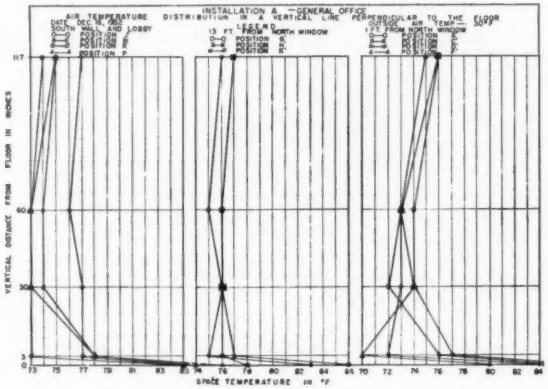


FIG. 12. AIR TEMPERATURE DISTRIBUTION IN A VERTICAL LINE PERPENDICULAR TO THE FLOOR FOR *Installation A*—GENERAL OFFICE

approximately 6 in. Beyond this point the rates of change for the 60 in. and 96 in. levels reverse and become equal at approximately 18 in. from the outside wall. This change in rate is due primarily to the convection effect of a non-uniform environment.

The main body of the air stream moving downward from the window makes contact with the floor at a distance greater than 1 ft from the wall; hence, the dotted portion of the curve for the 3 in. level is estimated.

Air temperature distribution in most rooms of the Minneapolis residence was determined for several weather conditions. Fig. 15 shows the temperature distribution at various locations in the living room, which due to the position of the exposed walls, is influenced more by the prevailing weather conditions. For all locations not influenced by convection effects from the windows and floor, the air temperatures are essentially the same.

FLOOR COVERINGS

The thermal resistance of the floor covering above the slab influences the surface temperature variation of the panel surface. The equalizing effect

depends on the thermal resistance of the floor covering. In addition the water temperature must be increased to provide the same temperature at the surface of the floor covering as would be required at the surface of a slab without any covering. Table 1 gives the average temperature drop through the various floor coverings used in the Minneapolis residence. Zone 1 employs a single thermostat located in the den and a single control valve. The average temperature of the panel water in Zone 1 is relatively constant. The result is that the space temperature in the bathroom runs several degrees higher when heating is required than the space temperature in the den. A marked degree of variation in the thermal resistance of the floor coverings makes it virtually impossible

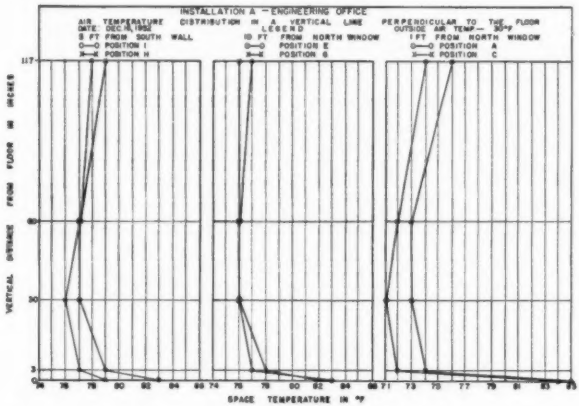


FIG. 13. AIR TEMPERATURE DISTRIBUTION IN A VERTICAL LINE PERPENDICULAR TO THE FLOOR FOR *Installation A*—ENGINEERING OFFICE

for a single thermostat to maintain uniform temperature levels, a condition usually considered desirable.

DISCUSSION

1. When a single coil is employed for heat supply to one or more separate enclosures within a space, floor covering of equal thermal resistance should be used in the separate enclosures. This does not preclude the use of one valve supplying a given inlet water temperature to two or more parallel coils in which an individual hand valve is used with each coil to regulate the rate of flow of water through the individual coil.

2. With two or more zones utilizing a common return header, in a control system employing continuous flow, when the control system calls for no energy supply to the zone being controlled, only return water is circulated through the coils. If other zones within the space require heat, the temperature of the water in the common header is of such magnitude as to cause a temperature drop through the zone in which return water only is recirculated. Consequently, after the individual controls have functioned to stop the supply of energy to the individual zone, a temperature drop through the coil occurs, thus supplying unwanted energy to the zone space. This effect

may be seen by observing any condition of the charts representing *Installation B*, in which the space temperature exceeds the set point while concurrently there exists a temperature drop of the water through the coils.

3. The throttling range of a control system employing a throttling or proportioning valve is described as the ratio

$$\frac{\text{thermostat change}}{\text{valve change}}.$$

A narrow throttling range describes the control system which produces a large valve action for a small space temperature change. The throttling range prescribed is selected on the characteristics of the response rate of the heating system and should be adjusted so as to result in a throttling or proportioning action. The limit of the

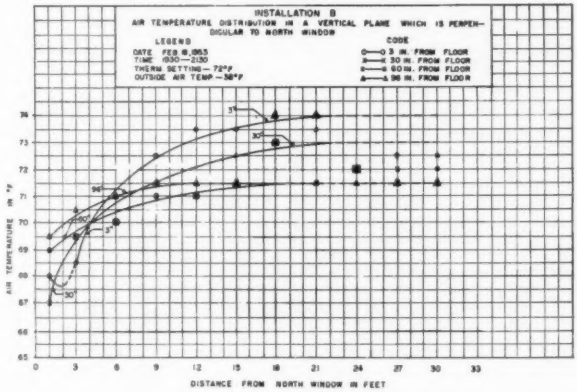


FIG. 14. AIR TEMPERATURE DISTRIBUTION IN A VERTICAL PLANE WHICH IS PERPENDICULAR TO THE NORTH WINDOW FOR *Installation B*

decreasing throttling range is effectively on-off control. Under conditions in which the rate of change in the heating load is greater than the response rate of the control system the operation with proportioning control is essentially two positions.

4. The on-off type valve operation provides a greater variation of panel water temperatures with less frequent operation. The frequency of the variation would be directly related to the heating load. The magnitude of the variation would depend on the method of boiler water temperature control. For constant boiler water temperature control, the magnitude of the variation of the panel surface temperature would vary with the heating load; and for variable boiler water temperature the magnitude would be an inverse function of the outside temperature.

5. The damping effect of a mixing valve eliminates the necessity for proportioning control of boiler water temperature. On-off control of the boiler around its control point is perfectly permissible provided the boiler control sensing bulb is located so as to sense the boiler water temperature. Location of the boiler control sensing bulb on the boiler outlet pipe some distance from the boiler tubes results in overheating of the boiler water when no boiler water is being circulated. This condition existed at the *Installation B*, through the 1951-1952 heating season, but was corrected for the 1952-1953 heating season. A comparison of the boiler water temperatures—curve 5—for the

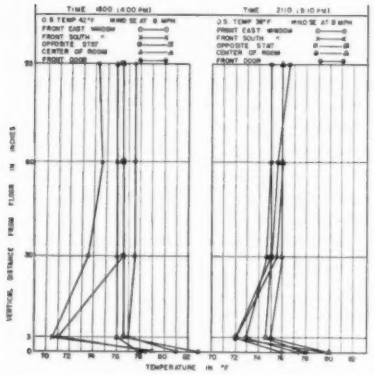


FIG. 15. AIR TEMPERATURE DISTRIBUTION FOR LIVING ROOM OF MINNEAPOLIS RESIDENCE ON MARCH 10, 1953

LEGEND

South Wall 30 percent Glass Area
East Wall 40 percent Glass Area
Floor Covering — 40 oz Foamtread and Heavy Carpeting
Ceiling Height — 8 ft

noon period of January 31, 1952 (Fig. 5) with the noon period of January 6, 1953, will illustrate the importance of proper location of the boiler control bulb to sense the true boiler water temperature.

6. The function of a space thermostat when used in conjunction with reset boiler water temperature is to sense the change in internal or space temperatures which would normally occur from conditions other than the change in outdoor temperature. Several periods are shown throughout the temperature trend charts in which an outside temperature change produced no change in the space temperature. If the boiler water temperature is reset with outdoor temperature, the thermostat functions as an auxiliary or secondary control element of the overall control system. If the boiler water temperature is constant, the thermostat functions as a primary control element of the overall control system. The frequency and magnitude of the varying space temperature for a constant heating load, would depend on the sensitivity and differential of the thermostat.

The complete field studies indicate that more desirable control is effected by adjusting the boiler water temperature with the outside temperature.

TABLE 1—EFFECTS OF FLOOR COVERING

ZONE	LOCATION	FLOOR COVERING	AVERAGE TEMPERATURE OF SLAB SURFACE	AVERAGE FLOOR SURFACE TEMPERATURE	AVERAGE TEMPERATURE DROP THROUGH FLOOR COVERING
1	Bathroom	Tile	86.0	84.0	2.0
	Den	40 oz combination rubber and fiber mat with heavy carpeting	86.0	75.0	11.0
2	Master Bedroom	13/16 in. Oak Parquet	88.0	79.5	8.5
3	Living Room	40 oz combination rubber and fiber mat with heavy carpeting	90.7	80.0	10.7
4	Kitchen	Linoleum	82.1	76.5	5.6

7. The effect of the thermal inertia of the panel slab is critical for any type of building where the exposed wall area is in large part glass, since the temperature lag through glass is virtually zero. If the glass area is exposed to solar energy the control problem is increased. Therefore, the addition and proper location of a solar compensator would, in part, alleviate the control problem associated with large glass area exposed to the sun.

For a floor panel heating system, the rate of response of the panel slab has more influence on the space temperature than the rate of response of the controls or the boiler.

8. As shown by the field studies using all the available control systems of each installation, space temperature is controlled much more effectively when the outside temperature is constant, and the heating load is approaching the design load at the system. The effectiveness of controlling a constant heating load decreases, however, when the capacity of the heating system becomes large in relation to the load.

9. The design of a floor panel heating system for a building located in an area having large heating loads should emphasize the concentration of thermal energy at the exposed walls, more especially at the windows, to offset the downdraft of cold air from the cold surfaces. Reduction of this convective effect improves the temperature distribution of the space.

10. The vertical location of the thermostat within the zone being controlled is not critical between the levels of 30 in. above the floor and 12 in. below the ceiling.

CONCLUSION

Considerable information has been obtained on floor panel heating systems in the field studies to date. Various design improvements are indicated from the results of the installations thus far studied. It is felt that, to warrant further studies of the control of floor panel heating systems, an installation in a controlled environment would be necessary.

ACKNOWLEDGMENTS

The authors wish to thank the Barber-Colman Co., Rockford, Ill., the Johnson Service Co., Milwaukee, Wis., and the Minneapolis-Honeywell Regulator Co., Minneapolis, Minn., for the controls which they provided, and the assistance and counsel they rendered.

DISCUSSION

E. L. SARTAIN, Urbana, Ill.: The authors included in the paper a discussion concerning the effects of carpeting. Here it is indicated that an average temperature difference across the carpeting of about 11 deg for a combination of carpeting and padding was experienced. Over what periods of time was this average taken, what was the outdoor temperature and the variation in outdoor temperature during this time?

C. F. KAYAN, New York, N. Y.: To what extent would the authors say the temperature of the water would have to be changed with a carpet in order to maintain the same conditions?

AUTHORS' CLOSURE (A. B. Algren): Mr. Sartain's question brings out an important point as to what is considered the surface temperature of the carpeting and how to determine this temperature.

For this phase of the project the instrumentation consisted of a series of 36 thermocouples mounted on a light wooden frame 12 in. wide and 24 in. in length. The carpeting was heavily napped and the tempeature recorded varied with the weighting of the frame to which the thermocouples were attached. The temperature at the end of the nap did not represent what we considered to be the effective surface temperature of the carpet as related to comfort.

The equipment was then weighted so as to determine a carpet surface temperature that would represent a comfort foot temperature. Therefore the temperature drop between the temperature of the concrete surface and what we considered the surface temperature of the carpet would be materially reduced over that when using the surface temperature of the nap.

Considering the comfort viewpoint the control of the space should be of such degree as to provide a comfort foot temperature when standing on the carpet.

These results were the average of readings taken over two 12-hr periods with the outdoor temperature varying between 20 and 30 F.

In answer to Professor Kayan's question, the temperature of the water with a carpeted surface would depend upon the insulation under the concrete slab. In this installation the L type insulation was used around the perimeter of the room. Three heat flow meters and thermocouples were installed in this area; one between the carpet padding and the top of the concrete slab, one under the concrete slab and one under the 2 in. of insulation.

Heat flow meters were also installed in an area where no insulation was used. The water temperature would be increased about 40 deg without insulation under the slab and approximately 10 to 15 deg with insulation. The reverse heat flow losses are of greater magnitude with carpeting and no insulation under the concrete slab, than with a bare concrete floor. This phase of floor panel installations requires further study and does not readily lend itself to a laboratory study, especially when no insulation is involved.

1501

No. 1501

COSTS OF OPERATING THE HEAT PUMP IN THE EQUITABLE BUILDING

By J. Donald Kroeker* and Ray C. Chewning**, Portland, Ore.

THE PERFORMANCE of the heat pump installation for year 'round air conditioning of the Equitable Building in Portland, Ore., has been observed for a period of one year. This paper discusses the results obtained with reference to energy costs for heating and cooling, and costs of maintenance, repair and operation.

Derivation of the cost of operation, segregated with regard to heating and cooling, of a heat pump system controlled automatically and providing heating and cooling simultaneously is complex and requires analysis and understanding of a large number of factors; namely, (1) the building, (2) use as affecting design, costs, and operation, (3) the heat pump system, (4) basic differences between a heat pump and a conventional system as affecting costs, and (5) heating and cooling load distribution.

Further considerations in providing a paper of usefulness require inclusion of comparisons of operating costs with those of conventional separate systems for heating and cooling, conversion of final cost figures to rate schedules in other cities, and recommendations for consideration of heat pumps for larger buildings, as may be deduced from the papers and experience on the Equitable Building heat pump.

The Building Described

While a general description of the building, its air conditioning system, and its heat pump are included herein, as necessary in discussion of costs, a more adequate understanding thereof will be attained by review of the previous papers. The first paper[1] described the system and considerations in selection of the design used in the system. The second[2] reported operating experience, heat balances, and thermal efficiencies found effective in one year's operation of the system. It had been planned to include cost data in the second paper, but accurate costs could not be determined at that time, because suitable recording electrical demand meters, necessary to determine increments of demand chargeable to heat pump operation, were not available for installation.

A 12-story office building with a basement and two penthouse levels, the building, Fig. 1, rises 200 ft above the street, is 200 × 100 ft in area at the

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[1] A Heat Pump in an Office Building, by J. Donald Kroeker and Ray C. Chewning (A.S.H.V.E. Transactions, Vol. 54, 1948, p. 221).

[2] Heat Pump Results in Equitable Building, by J. Donald Kroeker, Ray C. Chewning, and Charles E. Graham (A.S.H.V.E. Transactions, Vol. 55, 1949, p. 345).

Presented at the 60th Annual Meeting of The American Society of Heating and Ventilating Engineers, Houston, Tex., January 1954.

street level and 62 × 200 ft at the third floor level and above. It has a floor space of 212,000 sq ft (of which 148,711 sq ft is rentable space) and a volume of 2,275,000 cu ft. Architecturally, it includes many innovations. It is the first all-aluminum-clad office building in the United States. The windows are double glazed, extend from column to column, and are flush with the building face throughout. The external panes of the windows are of heat-absorbing type glass.

USE AFFECTS DESIGN AND COST

The Equitable Building is a rental office building throughout. This had a bearing on system selection. Since each square foot of rentable space is revenue-



FIG. 1. THE EQUITABLE BUILDING

producing, mechanical rooms on floors were eliminated. Since rental buildings may be subject to space rearrangement at any time, an all-air system was designed to eliminate the need for subsequent rearrangement of mechanical features and to provide means of effecting conservation and use of heat which would normally not be recoverable,[1, 2] as well as being required for cooling. Provision for any desirable space rearrangement required was obtained by zoning by exposures[1] rather than by rooms.

The habits of the occupants have a material effect on costs. The building is in use from 8 a.m. until 10 p.m.; hence the time of operation is longer than usual. The building's electrical demand, exclusive of the heat pumps, has peaks at approximately 11 a.m., 1 p.m., and 4 p.m. The first and third peaks are due to capacity elevator operation accommodating occupants in their indulgence of the American coffee habit. If heat pump peaks during these periods can be avoided, consequent demands chargeable to the operation of the heat pump are reduced.

AIR CONDITIONING SYSTEM

The building is completely air conditioned. A simplified diagram of the system is shown in Fig. 2. The heat pump which furnishes the heating and cooling is composed of four diffusion-type centrifugal, water-chilling machines, Fig. 3. Well water is the heat source. Wells not being pumped are used for disposal.

Referring to Fig. 2, the system consists generally of the following:

1. Twelve air conditioning units, comprised of fans and heating and cooling coils delivering air to plenums, from which air is mixed at the command of room thermostats in several zones per floor.
2. A recovery coil in the exhaust air.
3. Two preconditioning coils, each to supply ventilation air to several conditioning units.
4. Four heat pumps.
5. Other pumps illustrated.
6. A settling tank, to collect sand, if any, from well water.
7. Two warm wells and pumps, used for disposal during cooling.
8. One cool well and pump, used for disposal when heating is greater than cooling.

During periods when heating is greater than cooling the following operation is generally applicable:

1. The warm water supply and return system is a closed circuit in which water is circulated by the heating pump.
2. The warm well pump supplies well water to the settling tank pump.
3. The ST (settling tank) pump supplies water through valve W1 and the cooling pump, as required by thermostat T1, to the evaporator where heat is absorbed from it by chilling it. The heat absorbed is transferred to the refrigerant and becomes available for heating (see preceding section, paragraph 1).
4. Chilled water is supplied to the several cooling coils.
5. The discharge from the cooling coils passes through a recovery coil, absorbing heat from exhaust air, and thence to the preconditioning coils, where incoming ventilation air is heated by this waste chilled water.

6. The chilled water is then wasted to the casing of the cool well through valve S1. The amount wasted is equal to that injected by the ST pump, the remainder being recirculated by the cooling pump.

The temperature control for air conditioning and the heat pump is somewhat more complex than ordinary.[1] Heating and cooling are both available whenever the outside temperature is below 70 F. Well water from warm wells is injected into the evaporator circuit of the heat pump units whenever the heating load is the greater and from the cool well to the condenser circuit whenever the cooling load is greater. The system is completely automatic, except that, instead of being on a clock, switching to start the system is manual. Accordingly, well water

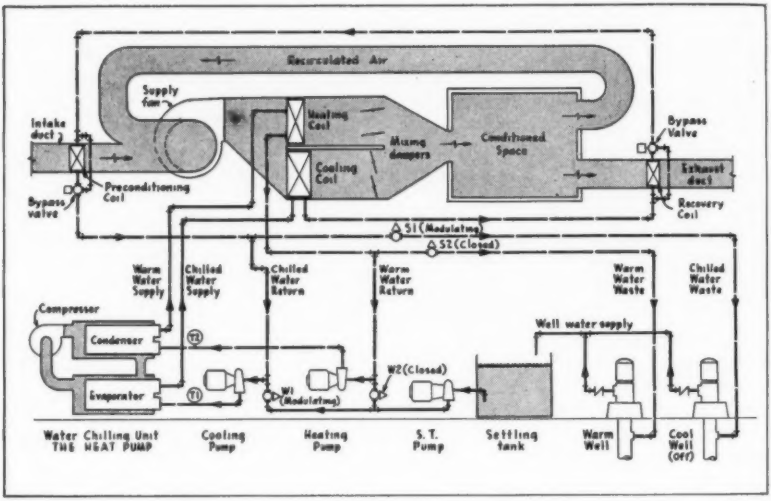


FIG. 2. FLOW DIAGRAM OF THE HEATING AND COOLING SYSTEM WITH VALVES POSITIONED FOR FLOW WHEN HEATING LOAD IS GREATER THAN COOLING

sources, operation to introduce well water, and capacity in operation are selected automatically.

BASIC HEAT PUMP CHARACTERISTICS

The significant difference between a heat pump and a conventional system, in which refrigeration is used for cooling and oil, gas, or other energy is used for heating, is the fact that, with a heat pump, heat may be obtained as a by-product anytime the system is in operation to obtain cooling. This factor, of great importance in the economical consideration of the heat pump as a heating system, was exploited in this heat pump installation.

In a building such as the Equitable Building, there are areas which require cooling even when the outside temperature drops to the design condition of 10 F. Usually, cooling loads occurring during normal heating periods are handled by the introduction of cool outside air.

In the Equitable Building this was not possible, because the space allotted to the mechanical equipment was such that it would have been impossible, without

reducing the rental space for each floor, to arrange the duct work to bring in the necessary outside air and to control it to prevent drafts from air which was too cool. Further, the cost of installation would have been considerably more to provide that type of system. Thus, if instead of a heat pump a conventional system with separate heating and cooling sources had been used in *this* building, the cooling equipment would have run every day of the year.

Reference to Fig. 4, developed in studies prepared for the second paper,[2] show that the minimum cooling load in the building is in the neighborhood of 125 Mbh. Therefore, comparing the heat pump system with a conventional type sys-

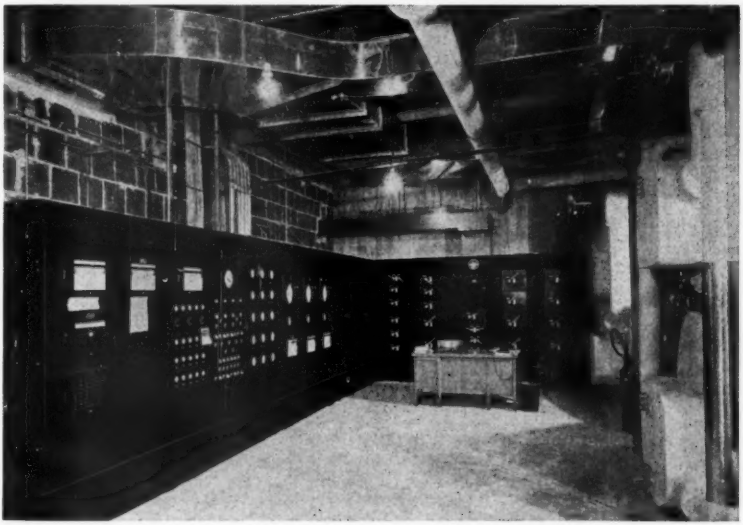


FIG. 3. MAIN MECHANICAL ROOM, SHOWING HEAT PUMPS AT RIGHT AND CONTROL PANELS WITH PERMANENT RECORDERS AND OPERATING CONTROL AT LEFT

tem, it becomes apparent that the refrigeration machinery would have operated to perform a cooling function even though the heating load was considerably in excess of the cooling load. Since heating is done with the heat rejected from this cooling load, this quantity of heat is a by-product of cooling and the energy required to obtain that quantity of heat can be charged to cooling. It is shown as *By-product Heating* in Fig. 4.

Fig. 4 shows that, at an outside temperature of 50 F, the cooling load becomes greater than the heating load. Since the data from which the curve was constructed were obtained from records for cloudy weather, the changeover point, 50 F, is high. Whenever solar heat is effective, the actual outside temperature at which the system changes from the condition of heating greater than cooling to cooling greater than heating is often as low as 42 F, as determined by a study of the demand meter records together with the Portland records of the U.S. Weather Bureau.

INSTRUMENTED FOR COST STUDY

Data on the electrical characteristics of the pump and the building were obtained by the use of recording, integrating, electrical demand meters, Fig. 5, installed by the local power company on a temporary panel to enable collection of data required for this cost study. The recording meters for the mechanical system, Fig. 3, are a permanent part of the system and were installed during

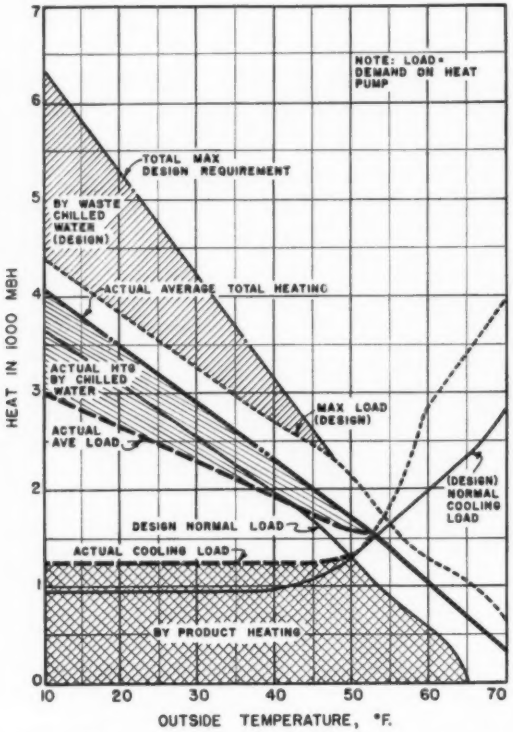


FIG. 4. HEATING AND COOLING REQUIREMENTS AND LOADS

construction.[2] Considerable data obtained from them were used in preparation of the paper on operating results.[2]

A recording demand meter for the building was also installed in the spring of 1949, when the meter became available. Operating results had revealed the importance of building demand characteristics in determining the simultaneous critical demand occurring in operation of the heat pump. Meter troubles resulted during the following heating seasons, so that the period from April 30, 1952, to May 1, 1953, was the first for which complete and unbroken records were available.

The separate recorders, registering maximum 30-min integrated demands, were installed for the heat pumps and auxiliaries as follows:

a. Water chillers, the heat pumps.
b. Settling tank pumps. (*ST pump* in Fig. 2).
c. Heating circulating pumps.
d. Cooling circulating pumps.
e. Warm well pump.
f. Cool well pump.
g. Building master meter.



FIG. 5. RECORDING ELECTRICAL DEMAND METERS INSTALLED TO ENABLE COST DETERMINATIONS

With these meters the following electrical demands and consumptions could be determined:

1. The building master meter could measure all energy used in the building.
2. Data on energy use for the building, exclusive of the heat pump and auxiliaries, could be obtained by subtracting the totals obtained from recorders *a* to *f*, inclusive, from readings of the building master meter.
3. Energy used by the heat pumps, settling tank pumps, heating and cooling circulating pumps and the warm and cool deep well pumps could be read directly.
4. Since the heating load is the greater when the warm well pump operates and cooling is the greater when no well water is used and when the cool well pump runs, periods chargeable to heating and to cooling could be determined from records of meters *e* and *f*.

The energy required for the fans in the several conditioning units could not be measured separately, because the wiring as installed makes segregation impractical. It is included in the general building energy figures.

METHODS OF ANALYSIS

The kilowatt and kilowatt-hour data for the building and the system were obtained from the demand meter charts after mounting them on a device specially-constructed to permit simultaneous examination of all seven meter records for any given month, Fig. 6. Hour-by-hour data were thus studied on a total of 670 ft of electric demand charts to obtain both electrical demands and energy use for the building and each group of equipment listed by the meters above.

Maximum monthly electrical demands were found by inspection. A check was then made to determine whether heating or cooling was the greater when the maximum demand for the month was established. This was done by inspection of the cool and warm well records.



FIG. 6. ELECTRICAL DEMAND CHARTS BEING ANALYZED BY GROUPS OF SEVEN

If it was found that cooling was greater when the maximum combined demand was set for the month, the breakdown was made as shown for June in the first half of Fig. 7; namely,

1. The maximum combined demand was recorded for the month at 2:30 p.m. on June 26 and consisted of building demand and demand of equipment to provide cooling.
2. The combined demand at a time when the demand for the building only was maximum was recorded at 3:30 p.m. on June 27.
3. The next bar in the graph shows the portion of the maximum combined demand for the month chargeable to cooling; namely, the difference between the maximum combined demand for the month and the maximum demand for the building only.

This procedure was used because cooling should be charged with only that portion of the maximum combined demand for the month which is in excess of the demand which would have been established if there were no cooling in the building.

If it was found that heating was the greater when the combined demand was maximum for the month, the breakdown was as shown for January in the second half of Fig. 7; namely,

1. The maximum combined demand was recorded at 11:00 a.m. on January 26.
2. The maximum combined demand occurring when cooling was greater than heating was recorded at 2:30 p.m. on January 12.

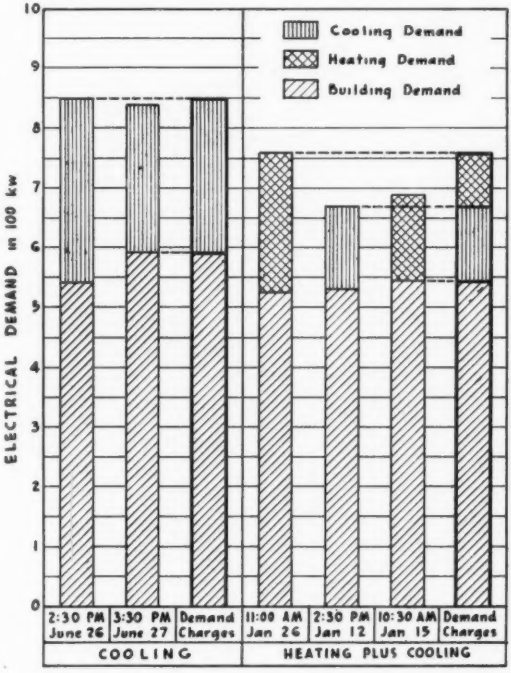


FIG. 7. BREAKDOWN OF DEMANDS CHARGEABLE TO HEATING, COOLING, AND BUILDING ONLY

3. The maximum demand for the building was recorded at 10:30 a.m. on January 15.
4. The last bar in the graph shows the distribution of demands to heating, to cooling, and to the building alone. The difference between the maximum combined demand when cooling was the greater and the maximum for the building alone, was charged to cooling. The difference between the maximum combined demand for the month and the maximum demand when cooling was the greater was charged to heating.

This procedure was used because the first requirement for consideration of the use of a heat pump is the decision to include cooling in the building design. The purpose of determining operating costs for the heat pump is to aid in the decision to use either the heat pump or a conventional cooling system with conventional heating for building air conditioning. The heat pump, then, should be charged only with that portion of the maximum demand for the month which is

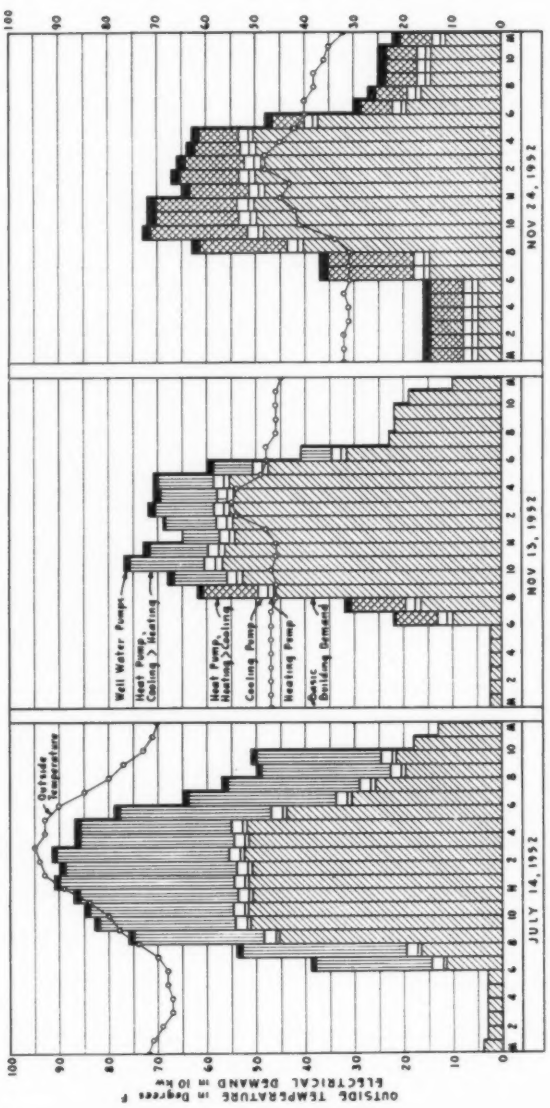


FIG. 8. MARCH OF HOURLY ELECTRICAL LOADS FOR THREE SELECTED DAYS

in excess of the demand which would be established by the conventional cooling system. The demand established by a heat pump when the cooling is greater than the heating is the same as would be established by a conventional cooling system. Again, the cooling demand should be only that portion of the maximum demand established when cooling is greater than heating which is in excess of the maximum demand which would have been established by the building if the building had no cooling system.

The determination of energy consumption was less complex, although much more laborious. The data were obtained by planimeter traverses of the recording demand meter charts of each of the meters previously listed.

ANALYSES DISCUSSED

The analyses resulted in reams of tabulations, particularly in study of the march of daily demands, converted by half-hours to consumption of electrical energy. These analyses are represented for three selected days in Fig. 8. The chart indicates demands established by the end of each hour shown. Hourly periods rather than the 30-min periods recorded and computed are shown for simplification.

The building demand shown includes the demand of the fans in the several air conditioning units, as discussed in a following section Total Air Conditioning Cost.

The demand for heat pump operation when the heating load is greatest includes about 125 Mbh of cooling. As the heat pump demand increases due to a load greater than that required for heating, as determined by the source of well water, the energy becomes chargeable to cooling. This is illustrated in Fig. 8 by changes between 9 a.m. and 10 a.m. of November 13. Heating during subsequent hours is the by-product of cooling.

The circulating pump load for heating and cooling, a load with minor variation, is chargeable to the type of service in effect, whether heating or cooling, and is not segregated in this chart. The same is true of the well water pumps, including deep well pumps and the settling tank pump, shown as ST pump in Fig. 2.

The first chart on Fig. 8 for July 14, was selected because it illustrates operation for a day in which outside temperatures reached the design dry-bulb temperature, 95 F.

Operation on a day with 47 F outside temperature, in which the system changed from operation chargeable to heating to operation chargeable to cooling, is shown in the second chart, that for November 13, 1952. The maximum demand for the month, chargeable to cooling, occurred between 10 and 11 a.m. The morning hours were overcast; the afternoon partly sunny.

To illustrate a day in which all energy use of the heat pump was chargeable to heating, the data for November 24, 1952 were selected. The day was overcast.

The chart illustrates a type of operation not discussed; namely, manual heating. At 6:15 p.m. of the previous day, as well as at 6:15 p.m. of the day shown, the operating engineer concluded that a cold night was in prospect and that continuous operation through the night would reduce the pick-up load on the following morning. Accordingly, he set certain equipment to run all night. This included one 70-ton heat pump, the heating and cooling pumps, fans on alternate floors, and, automatically, the settling tank pump and the warm well pump. By

doing so, he avoided a high pick-up load which would have probably resulted in a high demand chargeable to heating.

Had the temperature been 15 F or 20 F, he would probably have operated a 200-ton heat pump and all fans through the night. The manual operation illustrated is used in cold weather between consecutive days and over weekends and holidays. The effect of it is to increase the energy use chargeable to heating but to avoid chargeable demands.

DATA SUMMARIZED

A summary of electrical demands and energy use by months is given in Table 1. Segregation of operating time and electrical consumption to types of operation was necessary to enable determination of amounts of energy used for

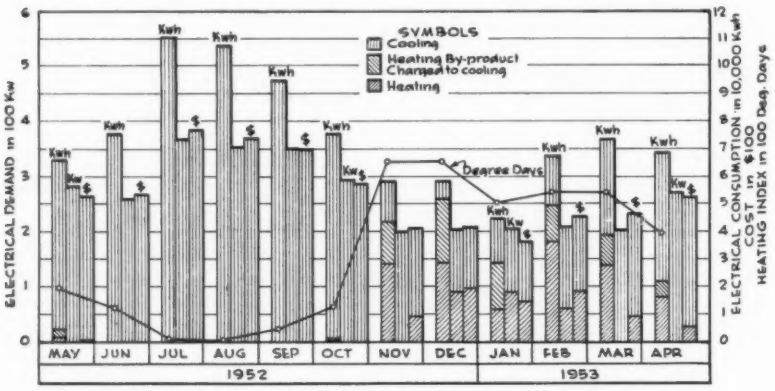


FIG. 9. MONTHLY ENERGY DEMAND AND USE AND COSTS

heating but chargeable to cooling as by-product heating, resulting in corrected values of kwhr for heating and cooling. By-product heating is considered the heat rejected during minimum cooling, Fig. 4.

The energy rejected to heating during minimum cooling was considered that effective during ideal transfer conditions; namely, a high chilled water temperature and a low condensing water temperature. In practice such conditions are not effective during applicable periods. However, they would be effective in a conventional system, in which it would be possible to do the minimum constant cooling under the ideal conditions mentioned.

Accordingly, to be comparable, the correction was made by determining the energy required by the heat pump, exclusive of auxiliaries, to produce the necessary cooling effect with a chilled water supply temperature of approximately 50 F and a leaving condensing water temperature of approximately 85 F, and multiplying this quantity by the hours of operation with heating greater than cooling.

Comparing the amount of energy used for heating with that produced as a by-product of cooling, it can be seen that, of a total heating requirement for the year of 243,801 kwhr, 89,370 kwhr or 37 percent is obtained as a by-product of cooling.

COST DERIVED

The rate schedule applicable to the computation of power cost is shown in Table 2. In calculating costs, kilowatt-hours were charged at the rate of 3.3 mills per kilowatt-hour, the last block in the schedule. The demand was charged at the full rate of $1.10 per kilowatt without any deduction. This is considered proper because air conditioning, whether by conventional methods or by a heat pump, is a load added to the building after application of the general necessary electrical uses, such as lighting, elevators, and other electrical uses standard in any building, whether air conditioned or not.

A break-down of electrical costs according to type of operation, *i.e.*, heating or cooling, and according to energy use, kwhr, and demand, kw, for each month

TABLE 1—ANALYSIS OF ELECTRICAL DEMANDS AND ENERGY USE

MONTH	HTG DEG DAYS	OPERATION						BY-PRODUCT[b] HEATING	CORRECTED CONSUMPTION (KWHR)		ELECTRIC DEMANDS[c]	
		Time (Hrs)			Consumption (Kwhr)							
		Htg	Man[a]	Clg	Htg	Man[a]	Clg		Htg	Clg	Htg	Clg
May	198	32	—	322	4,694	—	61,321	2,880	1,814	64,201	—	282
June	122	2	—	344	328	—	74,863	180	148	75,043	—	259
July	12	—	—	429	—	—	110,512	—	—	110,512	—	367
August	8	—	—	409	—	—	106,205	—	—	106,205	—	353
September	49	—	—	362	—	—	94,981	—	—	94,981	—	350
October	126	12	—	344	1,737	—	73,435	1,080	657	74,515	—	295
November	653	170	145	121	28,023	15,354	14,783	15,300	28,077	30,083	—	200
December	654	256	152	64	36,329	15,695	6,591	23,040	28,984	29,631	90	114
January	505	190	61	118	26,753	5,044	13,134	17,100	14,697	30,234	90	107
February	544	149	231	145	25,360	24,163	18,039	13,410	36,113	31,449	60	149
March	543	124	208	229	20,343	18,617	34,467	11,160	27,800	45,627	—	203
April	394	58	126	316	9,727	11,634	47,343	5,220	16,141	52,563	—	270
Total	3808	993	923	3203	153,294	90,507	655,674	89,370	154,431	745,044	240	2949

[a] Equipment on manual heating [b] Kwh [c] Kw

of the year from April 30, 1952, to May 1, 1953, is shown in Table 3, derived from Tables 1 and 2.

The monthly data on demands, energy use and costs are shown in Fig. 9. Energy consumption for heating, while generally following degree-days, departed from direct relationship in January and February due probably to sun effects and to more use of manual night time operation than may have been ideal for the purpose of its use. Costs paralleled demands more closely than they followed energy consumption.

The total cost due to demand charge for the year, Table 3, was $3,507.68, while the cost due to energy use is only $2,968.27. These figures demonstrate the fallacy of estimating electrical costs for heat pump systems on the basis of kwhr required to run the equipment.

The cost of energy per square foot of net rentable area, 148,711 sq ft, developed as 0.52 cents for heating and 3.85 cents for cooling, or a total for heating and cooling of 4.37.

While the kilowatt demand charge forms much the larger portion of the cost of air conditioning as a whole, it is approximately 1/3 the total cost of heating. This is due to the fact that only that amount of the heating demand charge

TABLE 2—ELECTRICAL MONTHLY RATES
(Schedule 32, Portland General Electric Co.)

	FROM	TO	COST
Demand Charge Dollars Per Kw	0	10 Kw	None
	Excess above	10	$1.10
Energy Charge Cents Per Kwhr	0	150 Kwhr	Cents 3.30
	150	500	2.20
	500	1,500	1.10
	1,500	15,000	0.88
	15,000	40,000	0.66
	40,000	100,000	0.44
	Excess over	100,000	0.33

which is in excess of the largest cooling demand charge occurring during any given month is charged to heating and is also due to the fact that the largest demands for heating energy generally occur at a time when the general building demand is very low. Heating exerts its heaviest demand during the early morning hours between 6 a.m. and 9 a.m. when the general building load is at a minimum. Conversely, the cooling demand occurs at a time when the building demand is at a peak. These factors are illustrated in Fig. 8.

TOTAL AIR CONDITIONING COSTS

Total costs of operation of the air conditioning system include items other than those discussed so far; namely, cost of fan operation, salaries for operating personnel, cost of maintenance and repair, and the cost of water treatment. These costs, together with segregation to heating and cooling, are shown in Table 4.

The cost of energy required for operation of the fans was not included in the analysis of the cost of operation of the heat pump, because the fans were not

TABLE 3—ANALYSIS OF ELECTRICAL COSTS FOR HEAT PUMP OPERATION
(Cost in Dollars)

	ENERGY USED			DEMAND OR KW			TOTAL		
	Heating	Cooling	Total	Heating	Cooling	Total	Heating	Cooling	Total
May '52	5.99	211.86	217.85	—	310.20	310.20	5.99	522.06	528.05
June	0.49	247.64	248.13	—	284.90	284.90	0.49	532.54	533.03
July	—	364.69	364.69	—	404.14	404.14	—	768.83	768.83
Aug	—	350.48	350.48	—	388.08	388.08	—	738.56	738.56
Sept	—	313.44	313.44	—	384.56	384.56	—	698.00	698.00
Oct	2.17	245.90	248.07	—	324.94	324.94	2.17	570.84	573.01
Nov	92.65	99.27	191.92	—	219.56	219.56	92.65	318.83	411.48
Dec	95.65	97.78	193.43	99.00	125.84	224.84	194.65	223.62	418.27
Jan '53	48.50	99.77	148.27	99.00	117.48	216.48	147.50	217.25	364.75
Feb	119.17	103.78	222.95	66.00	164.12	230.12	185.17	267.90	453.07
Mar	91.74	150.57	242.31	—	223.30	223.30	91.74	373.87	465.61
April	53.27	173.46	226.73	—	296.56	296.56	53.27	470.02	523.29
Total	509.63	2458.64	2968.27	264.00	3243.68	3507.68	773.63	5702.32	6475.95
Total Energy cost per sq ft of Rentable Space (cents)							0.52	3.85	4.37

considered a part of the basic heat pump system. However, in order to present a true picture of the complete cost of air conditioning, it was necessary to include the cost of energy supplied to the fan motors. As indicated, means of metering the power to the fans separately was not available. A good estimate of the power required for the fans was obtained by a knowledge of the types of fan operation possible and by use of the general building demand recorder. Three types of fan operation are possible:

1. Operation with all outside air intake and exhaust fans off when the building is unoccupied, for the purpose of maintaining minimum temperatures.
2. Operation with normal outside air and exhaust fans on.
3. Operation with 100 percent outside air and exhaust at temperatures of 60 F, Fig. 4, for the purpose of obtaining reduced operation of the heat pump.

The power required for the fans was determined from the general building meter by shutting off all uses of electricity in the building and reading the de-

TABLE 4—TOTAL COSTS[a] OF AIR CONDITIONING THE EQUITABLE BUILDING
(From Apr. 30, 1952 to May 1, 1953)

ITEM	DESCRIPTION	HEATING DOLLARS	COOLING DOLLARS	TOTAL	
				Dollars	Percent
1	Heating Energy	773.63	—	773.63	4.6
2	Cooling Energy	—	5,702.32	5,702.32	33.9
3	Fan Energy	161.50	1,961.64	2,123.14	12.6
4	Operators' Salaries	1,790.00[b]	5,370.00	7,160.00	42.5
5	Maintenance & Repair	136.00[b]	544.00	680.00	4.0
6	Water Treatment	400.00	—	400.00	2.4
7	Total	3,261.13	13,577.96	16,839.09	100.0
8	Total per NRA[c]	0.022	0.091	0.113	

[a] Does not include amortization of first costs.
[b] Breakdown as estimated by C. E. Graham, Chief Operating Engineer.
[c] Cost per year per square foot of Net Rentable Area.

mand made by the fans for each type of operation. The requirements were 53 kw for operation without outside air, 67 kw for normal operation, and 83 kw for 100 percent outside air operation. Since fan power requirements are quite constant, the knowledge of the demand and hours of operation were sufficient to make an accurate estimate of electrical use and costs.

The amount shown for total salaries of operating personnel is based upon one-half the total time of a chief operating engineer and two assistants. Many duties connected with the general building and the plumbing maintenance performed by these men were estimated as leaving one-half of the salaries chargeable to air conditioning.

Maintenance and repair costs, item 5, Table 4, include cost of materials or work performed by others, who were not operating men, on bearing lubrication, valves, refrigerant replacement, record charts, pump parts, motor repair, and filter service. Approximately 25 percent of this item is involved in filter service.

The total costs of energy are segregated with regard to heating and cooling in Table 3. Several methods of segregating costs, other than energy, for heating

or cooling could be justified, depending on the type of system with which comparison might be desired. The following methods are used:

1. Fan energy, item 3, Table 4 as segregated was 92.4 percent to cooling for three reasons; (1) cooling was the basic scheme, (2) cooling occured at all times, except during manual heating, and (3) the air systems to which the fans are basic were selected primarily because cooling was a prime requirement. Fan energy was thus charged to heating for periods of manual heating.

2. Operators' salaries and cost of maintenance and repair, items 4 and 5, Table 4, could not be apportioned to heating and cooling on the basis of demands or energy consumptions. Accordingly, the breakdown represents values justified by the Chief Operating Engineer.

3. Water treatment, item 6, Table 4, would have been unnecessary if water only from the cool well had been used;[1] accordingly the cost is charged to heating.

Total costs per square foot of net rentable area developed as 2.1 cents for heating and 9.2 cents for cooling, or a total of 11.3 cents for air conditioning.

LIMITATIONS OF DATA

The accuracy of data presented was affected by several factors beyond control.

The recorder chart multiplier for the master meter was 1000 and the reading accurate to only two significant figures. It was therefore, possible to obtain combined electrical demand values to only the nearest 10 kw.

The values for energy consumption, determined by planimeter from the charts, were not identical with metered values, although they were within a few percent of them. The margin of error occurred in variations of chart base lines and record conformations.

Heating and cooling pump loads were charged to the heat pump operation. While this is somewhat unfair, at least in comparisons with conventional systems, these loads could not be segregated with accuracy.

The amount of sun effect, having a significant bearing on loads in a building with maximum glass areas, could not be satisfactorily indexed to relate to the data. Weather Bureau designations available for downtown Portland are unsuitable for the purpose. A cirrus cloud cover, which may be designated as 10/10 sky cover, may transmit as much as 80 percent of normal solar intensities. Other conditions given as 7/10 sky cover may be 70 percent scattered clouds or 70 percent uniform cumulus cloud cover with 30 percent clear skies, variations with significant effect on air conditioning requirements. Accordingly, loads did not follow temperatures as expected. See Fig. 8.

COMPARISONS WITH OTHER SYSTEMS

Discussions of heat pumps and their installation, operation, and costs naturally develop into comparisons with conventional systems of air conditioning buildings. The latter include heating systems using district steam or heating plants contained in buildings, together with cooling systems. Table 5 presents such a comparison. It is based on the all-air system installed in the Equitable Building using steam from a district steam system or No. 5 fuel oil to provide hot water circulation to heating coils in the conditioning units and the preconditioning coils, instead of condenser water.

First costs of conventional systems are not estimated. The first cost of a

TABLE 5—COMPARISONS OF COSTS OF ENERGY OF A HEAT PUMP SYSTEM AND A CONVENTIONAL SYSTEM IN THE EQUITABLE BUILDING

HEATING ENERGY	HEATING COST		COOLING COST DOLLARS	TOTAL COST DOLLARS
	DOLLARS	%		
The Heat Pump System.....	773.63	100	5,702.32	6,475.95
District Steam[a].....	4,930.00	638	5,702.32	10,632.32
No. 5 Fuel Oil[a].....	3,430.00	444	5,702.32	9,132.33

[a] With separate cooling system as installed.

plant using No. 5 fuel oil would include the cost of a boiler plant within the building and of a required chimney, and the annual costs would include annual value of the space required for boilers.

Annual costs are based on the heating and cooling loads developed for the Equitable Building, the local district steam rates, averaging $0.85 per 1000 lb, and No. 5 oil at $2.71 per barrel. These values are based on comparisons of energy cost only derived from average temperature data, resulting loads under overcast sky conditions from Fig. 4; and heat balance curves developed for the second paper.[2]

Using district steam, which is available and would have been used had a heat pump system not been adopted, the total cost of energy for air conditioning for the year would have been 65 percent greater. Table 5 also shows that the cost of energy for cooling was only 17 percent greater than the cost of district steam for heating only. The cost of energy for heating and cooling the Equitable Building was only 31 percent greater than the cost of district steam would have been for heating only.

Portland district steam rates are relatively low. If estimates on district steam use for the Equitable load were based on rates in other cities, instead of the average rate of $0.85 per 1000 lb, they would be developed at $1.46 for New York, $1.15 for St. Louis, $1.52 for Seattle, $1.64 for Chicago, and $1.43 for Minneapolis, the average rates given in the 1952 report of the Operating Statistics committee of the *National District Heating Association.*

COMPARISON WITH OTHER CITIES

Electrical rates for the Pacific Northwest are generally understood as exceptionally low. Table 6 is a conversion of costs to rates of other cities con-

TABLE 6—COMPARISONS OF COSTS OF ENERGY AT ELECTRICAL RATES OF OTHER CITIES

CITY	SCHEDULE[a]	HEATING DOLLARS	COOLING DOLLARS	TOTAL	
				Dollars	Percent
Portland		773.63	5,702.32	6,475.95	100
Seattle	Commercial	1,256.58	8,374.83	9,631.41	148
San Francisco	A-13	1,481.15	8,862.44	10,343.59	160
St. Louis	Large L & P	1,415.53	9,222.65	10,638.18	164
Chicago	Large Commercial	1,132.45	8,529.18	9,661.63	149
Minneapolis	Large Commercial	1,963.53	12,559.37	14,522.90	224

[a] From National Rate Book of the Federal Power Commission.

sidered applicable; namely, San Francisco, St. Louis, Chicago, Minneapolis, and Seattle, as published in the National Rate Book of the Federal Power Commission. The rates on demand charges in other cities are understood to be based on the maximum 15-min integrated demand occurring in a month instead of the 30-min basis used in Portland. The 30-min demands of the recorder charts were considered as 15-min demands in computing values for Table 4. The comparisons are based on quantities developed for the Equitable Building, Table 1, without attempted conversion by degree-day ratios to these cities. Approximate conversions of heating costs by heating degree-day ratios, if made, should be considered accurate to no more than ± 15 percent.

Adaptation of a similar heat pump system or any large heat pump system to another location is attended by consideration of other physical factors requiring individual analysis not considered here. These include availability of well water, practicability of disposing to a stratum not being pumped, sewer charges if disposal to ground is impracticable, and water charges if well water is not available at reasonable cost. Use of city water as a heat source generally will rule out economic feasibility.

SUMMARY

Methods of analysis of operating heat pumps are complex and depart widely from usual considerations. Heat pump operation cannot be evaluated in the normal manner applicable to conventional systems. Few heat pump installations will warrant equipment and costs of analyses required.

Significant cost data developed and applicable to the Equitable Building, probably exclusively, as disclosed in one year's operation are as follows:

1. Heating as a by-product of cooling represented 37 percent of total heating provided.

2. The cost of energy for the year per square foot of net rentable area totalled 0.52 cents for heating and 3.85 cents for cooling.

3. The total cost of operating the system for the year totalled 2.1 cents per square foot of rentable area for heating and 9.2 cents for cooling.

4. The cost of cooling was 17 percent greater than heating would have been if heating alone had been accomplished separately with district steam.

5. Heating with district steam would have cost 6.38 times as much as that provided with the heat pump as applied.

6. If district steam had been used, the total air conditioning cost would have been 65 per cent greater.

7. While Portland electric and district steam rates are comparatively low, it appears from percentages in Table 4 and from percentages calculated from average district steam rates for several cities given in the section, Comparisons with Other Systems, that the increase in cost of steam is higher than the increase in cost of electricity. Accordingly, the use of a heat pump such as that in the Equitable Building would appear more favorable in these cities than in Portland.

The most important fact revealed in this study of costs of operation of the Equitable Building is that, while a heat pump might not be economical when installed for the purpose of heating alone, its use in conjunction with cooling can save thousands of dollars per year as compared to a conventional heating and cooling system in a building as large as the Equitable Building.

Because the heat pump finds economic application as the means of heating only when conditions are favorable and a cooling system is desired, the determination of heating cost should be made on the basis of a system performing a comparable function. Demands should be computed as discussed and illustrated in Fig. 7.

Because it produces both heating and cooling from the energy required for cooling alone, the quantity of heat produced as a by-product of cooling must be considered.

The importance of flexible automatic control, and intelligent operation are great in the maintenance of economical operation.

The greater the emphasis placed upon demand charges in the electrical rate under which the heat pump must operate, the more economical will heating with a heat pump be in an air conditioning system.

In any large heat pump installation, the permanent installation of recording demand meters will probably be economically justifiable as an aid to the operating engineer in avoiding demand peaks chargeable to heating.

ACKNOWLEDGMENTS

Grateful acknowledgments are made to the following for assistance in obtaining the data reported herein: The Portland General Electric Co. for loan and monitoring of the recording demand meters; Messrs. Sharp, Cadigan, Wiitala, C. W. Brissenden, and A. N. Hoss of the power company for assistance with analyses; and C. E. Graham, chief operating engineer of the building, for daily monitoring of the recorders.

DISCUSSION

E. H. HATTERSLEY, Fort Wayne, Ind.: The authors state that the heat pump system is designed primarily to be used in connection with air units, for the purpose of space conservation and flexibility of building arrangement, as it might apply to office space arrangement. Is the heat pump practical for a split system in a factory building, or a factory office building, or an industrial building; with the use of circulating hot water in convectors or baseboard radiation, and using circulated air only as an auxiliary for summer cooling?

E. R. LOKEY, Portland, Oreg.: I would like to know what the cost of the survey was and how the moneys for the survey were secured.

H. E. ZIEL, Detroit, Mich.: In this air-handling system was the air change rate the same on the heating cycle as on the cooling cycle?

R. G. VANDERWEIL, Boston, Mass.: I want to express my thanks for this paper which is most instructive in conjunction with a general study of the heat pump, where so little operating information is available to the engineer.

I note in the presentation of this paper that the slide for Fig. 4 does not have the area called *Actual Heating by Chilled Water* as in the paper. Can the authors explain this discrepancy?

Concerning electric service charges, what is (1) the amount of the demand charge, and (2) the base for the hourly charge?

A final question was in connection with Table 5, Cost Analysis. If I properly understand the paper, any amount that may have been spent while refrigeration was required, was fully charged to cooling and not at all applied to heating. I wonder

whether it would not be fairer to charge, say, 50 percent to heating and the remainder to cooling?

AUTHOR'S CLOSURE (J. D. Kroeker): In reply to Mr. Hattersley concerning the use of the heat pump in systems, split systems and others, we feel that adaptability of the heat pump is almost infinite, within its temperature limitations. In one installation as large as the Equitable Building, that of the Oregonian Publishing Company, a split system was used.* The other side of the system was panels, receiving 135 F water during design conditions.

The chief advantage of the heat pump is its ability to conserve heat normally wasted and to make use of it to reduce operating costs.

In connection with Mr. Lokey's question, very simply stated, the cost of developing the paper on costs, including the study involved, was five and one-half times the cost of energy for heating the Equitable building for one year.

He also wanted to know where we got the money to spend on this study. I can say that in our work we have benefited from those who have gone before and the literature that they have left, and we thought that the least of our obligation was to provide information we might have that seemed desired by others.

In response to Mr. Ziel's question the amount of air circulated summer and winter is identical. It has resulted in no problems. Normally you would use considerably less air circulation in the wintertime, but, in a draftlessly arranged job, high air circulation rates during heating make no problems and they do give the advantage of handling large quantities for heat recovery.

Mr. Vanderweil had several questions. A discrepancy between my simplified graph and the one in the paper was pointed out, which exists not as a discrepancy but as a difference in interpretation. The graph in the paper is rather complex because it shows two things: (1) the design hourly values, and (2) the values found existing as 24-hr averages, not hourly values. The latter were obtained as a result of studying the operation of the system, as reported in 1949.† The upper two shaded areas in Fig. 4 of the paper are identical in designation or substance but on 2 different bases. Heat absorbed from waste chilled water on a design basis is shown in the upper shaded area. The lower one, the only one shown on the slide, represents actual values on a 24-hr basis.

The power rates are given in the paper: namely, ten kilowatts of demand without charge and excess above ten charged at the rate of $1.10 per kilowatt. The energy rate drops down to 3.3 mills per Kwhr in the quantities used in this building.

Mr. Vanderweil's suggestion with respect to Table 5 in the distribution of charges for heating may be another person's approach. Ours was for our climate, for the first building in our City ever to be cooled as a whole, and for this particular structure. Cooling was the basic requirement set by the owner. As a matter of fact, in the initial stage of planning the building he said, "I will erect an office building in Portland only providing I completely air-condition it. It must be cooled." Since the cooling requirement was set as basic as against heating, segregations shown in Fig. 7 seemed the proper approach.

Cooling is charged for the full requirement *only when* the cooling load is *greater* than heating.

Analyses of cost of operation of a heat pump must be related to normal applications of heating and cooling. If a conventional system had been used, heating would have been supplied by oil or district steam and cooling would have been totally charged to cooling. The paper enlarges further on the principles of segregation of the charges to heating and to cooling, as applicable to this installation.

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* Heat Pump Application to a Newspaper Plant, by J. Donald Kroeker, John H. Bonebrake and James A. Melvin (A.S.H.V.E. TRANSACTIONS, Vol. 57, 1951, p. 467).

† See Reference 2 of the paper.


1502

No. 1502

AVAILABILITY AND UTILIZATION OF SOLAR ENERGY†

By R. C. JORDAN* AND J. L. THRELKELD**, MINNEAPOLIS, MINN.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the Department of Mechanical Engineering of the University of Minnesota††

Part I—Solar Energy Availability for Heating in the United States

FOR THE PAST 500 million years petroleum deposits have been accumulating on the earth and for 250 million years coal has been accumulating. Man, a comparative latecomer, learned to harness this energy only during recent years. Over 85 percent of all the fossil fuels used have been consumed since 1900. Probably the best estimates[1] of the life expectancy of fossil fuel reserves in the United States vary from a minimum of about 75 years to a maximum of 250 years, an average of about 160 years. Although such figures are difficult to determine with accuracy and involve changing rates of fuel production and changing rates of demand, the order of magnitude is correct and the fact remains that sources of stored fuel are rapidly running out. Within another century natural petroleum deposits will be rare, 80 percent of liquid fuel will probably come from coal, and natural gas will be scarce. In the foreseeable future new sources of energy must be tapped or present increasing standards of living, so dependent upon energy and power, will degenerate.

With negligible exceptions, all energy sources originate from the sun. Those such as hydroelectric power and the burning of vegetation and trees for fuel are current and renewable. The greatest sources of energy for power or heating, however, are the fossil fuels, such as, petroleum, natural gas, coal, tar sands, and similar deposits. In effect, then, the sun is the only real source of energy, and the potentialities of its direct utilization dwarf all of the major energy sources available. If all of the earth's estimated coal, petroleum, lignite,

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† This paper is divided into three parts concerning (1) Availability of Solar Energy, (2) Utilization of Solar Energy, and (3) Design and Economics of Heat Pump Systems Using Solar Energy as a Heat Source. The three parts were presented as one paper and discussions follow the final part on page 231. A Bibliography appears on p. 229.

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†† This research work has been carried on under the guidance of the A.S.H.V.E. Technical Advisory Committee on the Heat Pump and will be used in partial fulfillment of the requirements of the doctor of philosophy degree of J. L. Threlkeld.

[1] Exponent numerals refer to References.

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natural gas and other fossil deposits ever likely to be produced plus the fuel potential of all timber were stockpiled, the total energy represented would be equivalent only to a three-day supply of the solar energy intercepted by the earth from the sun.

Unfortunately, solar conversion systems for the translation of the sun's energy to useful electrical or mechanical energy are still inefficient. The efficiency of utilization of solar energy in the process of photosynthesis is in the order of 0.1 to 0.5 percent. The overall efficiency of photoelectric cells, capable of direct conversion of light to electrical energy, approximates 1 percent. Thermoelectric conversion of heat to electricity by means of thermocouples has attained an encouraging 7 percent efficiency. Swiss solar water heating systems for the collection of solar energy and its storage in the form of useful heat are said to have attained peak collection efficiencies of 36 percent and average monthly efficiencies of 10 percent in March and August. The Dover, Mass., solar house designed for solar collection and heating has successfully attained collection efficiencies as high as 63 percent by contributions directly through south fenestration and by south-wall solar collectors. The direct collection of solar energy as heat is, at this time, much more efficient than its conversion to other forms of energy, and the first sizeable engineering contributions are likely to lie in this direction. Fortunately, tremendous contributions are potentially achievable, since nearly a third of the total energy requirements of the United States are consumed in the heating of homes and buildings. This figure is increased to almost 50 percent when expanded to include the world.

The temperature of the solar energy collected must be high if the energy is to be converted to power, but can be comparatively low if it is to be used for space heating. Further, the efficiency of collection and storage is low at high temperature but, in general, increases as the temperature decreases. For direct space heating, the collection temperature must be higher than the temperature of the rooms which it heats, but if aided by a booster or heat pump it can be lower than the temperature of the heated space. No advantage accrues, however, unless the temperature of the solar energy collected is higher than that of normally available heat sources for heat pumps.

Potentially, then, solar energy may be used for the direct production of power, it may be used directly for space heating, or it may be used indirectly as a partial or total heat source for heat pump systems. The conversion of solar energy to electrical or mechanical power is both possible and eventually probable. C. G. Abbot[2] of the Smithsonian Institution, associated for many years with solar energy utilization studies, believes 15 percent of intercepted solar energy may be usable for power. He predicts that Mexico, because of geographical position, could supply 10 trillion horspower hours per year or the equivalent of all the power annually consumed in the United States. H. C. Hottel of Massachusetts Institute of Technology believes solar power systems located in the Southwest could produce 50 hp per acre or about 14 trillion horsepower hours per year with energy collected over an area of about 40 percent of the state of New Mexico.

The purpose of this paper is to discuss the availability of solar radiation in the United States. The maximum radiation is available outside the atmosphere. As radiation passes through the atmosphere, it is split into three parts: (1) one part is turned aside from the direct beam and scattered in practically all direc-

tions; (2) another part is absorbed and changed into heat energy; and (3) the remainder is transmitted to the earth's surface unchanged in wavelength. During cloudless days 50 percent or more of the radiation available outside the atmosphere may be depleted, and during extremely cloudy days practically all may be depleted. Rational theoretical methods are known for calculating the irradiation of a surface outside the atmosphere and on the earth during cloudless days. However, because of the complicating effects of clouds, no such theoretical procedure is known for calculating the average solar energy available at the earth's surface during all days, and recourse must be made to experimental data.

DIRECT SOLAR RADIATION AVAILABLE OUTSIDE THE ATMOSPHERE

Position of Earth Relative to Sun: The location of any point on the earth relative to the sun is continually changing due to the two principal motions of the earth: (1) rotation about its axis, and (2) revolution about the sun. The axis of rotation of the earth is tilted 23.5 deg with respect to the plane of

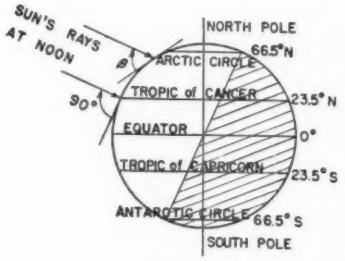


FIG. 1. POSITION OF EARTH RELATIVE TO NOON SUN'S RAYS ON JUNE 21

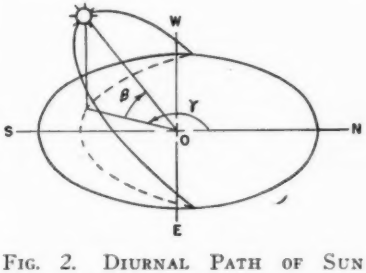


FIG. 2. DIURNAL PATH OF SUN ILLUSTRATING SOLAR ALTITUDE AND AZIMUTH ANGLES

its orbit about the sun. Fig. 1 shows the position of the earth relative to the noon rays of the sun on June 21 (the summer solstice). At this time the noon rays are normal to the earth's surface at the Tropic of Cancer. Regions south of the Antarctic Circle are enshrouded in darkness 24 hr per day, and there is continuous daylight north of the Arctic Circle. The reverse is true at the time of the winter solstice (December 21) when the noon sun's rays are normal to the Tropic of Capricorn. At the time of the equinoxes (March 21 and September 21) the noon sun's rays are normal to the equator.

To an observer on the earth, the earth is fixed and the sun moves across the sky following the path of a circular arc from horizon to horizon. Fig. 2 shows such a diurnal path and describes two important angles involved in solar radiation calculations. These angles are the solar altitude and solar azimuth. The solar altitude β is defined as the angle in a vertical plane between the sun's rays and the horizontal. The solar azimuth γ is defined as the angle measured in a horizontal plane from north to the horizontal projection of the sun's rays. These angles vary continuously from sunrise to sunset but are symmetrical with respect to the north-south line.

The position of the sun relative to the earth at any time of day may be established if its declination, altitude and azimuth angles are known. The declination of the sun which is the angular distance of the sun north or south of the celestial equator is given in the Nautical Almanac.[3] The sun's altitude and azimuth are given in U. S. Hydrographic Office Tables 214.[4] In order to use the tables, the latitude, local hour angle, and declination must be known. The local hour angle is expressed in degrees (0 deg for 12 noon, 15 deg for 11 a.m. and 1 p.m., 30 deg for 10 a.m. and 2 p.m., etc.).

The Solar Constant: The intensity of solar radiation at normal incidence, at the outer limit of the atmosphere, and at the mean solar distance is known as the *solar constant.* During the past hundred years numerous determinations have been made of its value. The most accurate of these were made since 1905 by Abbot[5] and Fowle. At the present time the most accurate determinations and corrections[6] fix the solar constant as 1.896 g-cal per (sq cm) (min) or 419.40 Btu per (hr) (sq ft). The earth's orbit is slightly elliptical with the sun

TABLE 1—RATIO OF RADIATION INTENSITY AT NORMAL INCIDENCE OUTSIDE EARTH'S ATMOSPHERE TO SOLAR CONSTANT

MONTH	DAY OF MONTH 1	DAY OF MONTH 8	DAY OF MONTH 15	DAY OF MONTH 22
January	1.0335	1.0325	1.0315	1.0300
February	1.0288	1.0263	1.0235	1.0207
March	1.0173	1.0140	1.0103	1.0057
April	1.0009	0.9963	0.9913	0.9875
May	0.9841	0.9792	0.9757	0.9727
June	0.9714	0.9692	0.9680	0.9670
July	0.9666	0.9670	0.9680	0.9692
August	0.9709	0.9727	0.9757	0.9785
September	0.9828	0.9862	0.9898	0.9945
October	0.9995	1.0042	1.0087	1.0133
November	1.0164	1.0207	1.0238	1.0267
December	1.0288	1.0305	1.0318	1.0327

located off-center of the ellipse, resulting in the earth being about 3½ percent closer to the sun on January 1 than on July 1. As a result, there is about 7 percent greater radiation outside the atmosphere on January 1 than on July 1 which is apparently a contributing factor to the more severe winters of the southern hemisphere. Table 1, based upon the data of Fowle,[7] presents factors by which the solar constant should be multiplied to give normal incidence radiation outside the atmosphere for various days of the year.

Radiation Intensity Outside the Atmosphere Incident upon Various Surfaces: The component of the sun's rays normal to a horizontal surface is at any time of day (see Fig. 2):

$$I_H = I_n \sin \beta \quad \ldots \ldots \ldots \ldots \ldots \quad (1)$$

where

I_H = radiation incident upon a horizontal surface outside the atmosphere, Btu per (hour) (square foot).
I_n = radiation intensity at normal incidence outside the atmosphere, Btu per (hour) (square foot).
β = solar altitude angle.

The component of the sun's rays normal to a south-facing tilted surface is at any time of day (see Fig. 3):

$$I_{ST} = I_N (\sin \beta \sin \phi + \cos \beta \cos \alpha \cos \phi) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where new quantities are defined as

I_{ST} = radiation incident upon a south-facing tilted surface outside the atmosphere, Btu per (hour) (square foot).
ϕ = angular tilt from vertical position.
α = wall-solar azimuth angle = 180 deg $-$ γ.
γ = solar azimuth angle.

For the case of a south-facing vertical surface, $\phi = 0$, and the incident radiation I_s is given by

$$I_s = I_n \cos \beta \cos \alpha \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Daily Total Radiation Outside the Atmosphere Incident upon Various Surfaces: Hourly rates of radiation throughout a day may be calculated by Equations 1, 2, and 3. Sample results for horizontal and south-facing vertical surfaces for January 15 and July 15 at 42 deg north latitude are shown in Table 2. Once hourly rates are known, the total radiation incident upon the surface during the day may be found by summing up the hourly rates using Simpson's rule. Such calculations were carried out at weekly intervals throughout the year for 30, 36, 42, and 48 deg north latitude for a horizontal surface, a south-facing vertical surface, and a south-facing surface tilted such that the surface is normal to the sun's rays on January 15. This results in a tilt angle from the horizontal approximately equal to the latitude plus 21 deg.

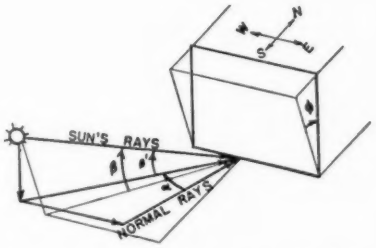


FIG. 3. RELATION OF SUN'S RAYS TO SOUTH-FACING SURFACE

The curves for daily total solar radiation incident upon horizontal and south-facing vertical surfaces outside the atmosphere for the various latitudes are shown in Fig. 4. In Fig. 5, curves of the ratio of the radiation incident upon a south-facing tilted surface (tilt angle from horizontal equal to latitude plus 21 deg) to that incident upon a south-facing vertical surface are shown.

Fig. 4 reveals that a horizontal surface intercepts a minimum of solar energy in winter and a maximum in summer, and that the reverse is true for a south-facing vertical surface. Fig. 5 indicates that little advantage results in northern latitudes by tilting a south-facing surface.

Since the curves of Figs. 4 and 5 are limited to solar energy reception outside the atmosphere, practical interpretation of the results in terms of reception of solar radiation at the earth's surface is also limited. However, the superiority of a south-facing vertical surface over a horizontal surface in the winter interception of solar energy is evident. In addition, south-facing vertical surfaces are more desirable in summer, since far less solar energy is incident upon them than upon horizontal surfaces at this time of year. The value of a south-facing tilted solar collector in relation to a south-facing vertical collector is questionable. There may be some justification for its use in southern latitudes, but in the more northerly latitudes, little additional solar radiation is incident in winter and a considerably greater amount is incident in summer when it is undesirable.

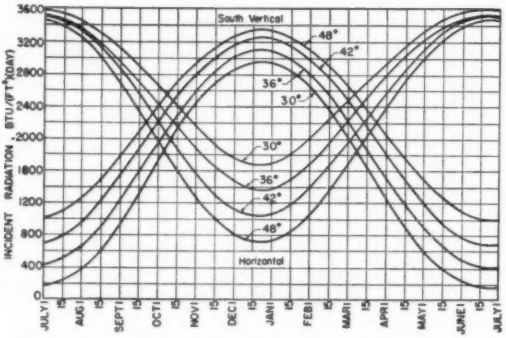


FIG. 4. DAILY TOTAL SOLAR RADIATION OUTSIDE THE ATMOSPHERE INCIDENT UPON A HORIZONTAL SURFACE AND UPON A SOUTH-FACING VERTICAL SURFACE AT VARIOUS NORTH LATITUDES

SOLAR AND SKY RADIATION AVAILABLE AT EARTH'S SURFACE DURING CLOUDLESS DAYS

Depletion of Solar Radiation by the Atmosphere: The rate at which direct solar radiation is received by a particularly oriented surface on the earth during cloudless days depends upon (1) radiation intensity at normal incidence outside the atmosphere, (2) solar altitude and azimuth angles as determined by latitude, time of day and time of year, and (3) depletion of radiation by the atmosphere.

Numerous investigators have studied the problem of atmospheric depletion. They have found that scattering of solar radiation is caused by molecules of dry air and water vapor and by dust particles. The principal constituents of the atmosphere which absorb solar radiation are ozone and water vapor, but this absorption is selective, *i.e.*, it depends upon the radiation wavelength. This is also true for scattering. Thus, the spectral distribution of radiation, both outside the atmosphere and at the earth's surface, must be considered.

One of the best papers on the theoretical and experimental aspects of atmospheric depletion is that of Parry Moon.[8] He has calculated spectral irradiation

curves for a surface perpendicular to the sun's rays at the earth's surface during cloudless days. The A.S.H.V.E. has taken these results as a standard for clear atmospheres and for typical summer conditions.

F. W. Hutchinson and W. P. Chapman[9] have applied Moon's theory to winter-time conditions. They assumed conditions of 760 mm of mercury barometric pressure, water vapor content corresponding to 33 F dew-point temperature, 300 dust particles per cc, and 2.8 mm of ozone. The results of their calculations are shown in Fig. 6 where direct solar radiation at normal incidence is presented as a function of solar altitude angle. This curve is for the month of January only. The extension to other winter months is made by multiplying the January values

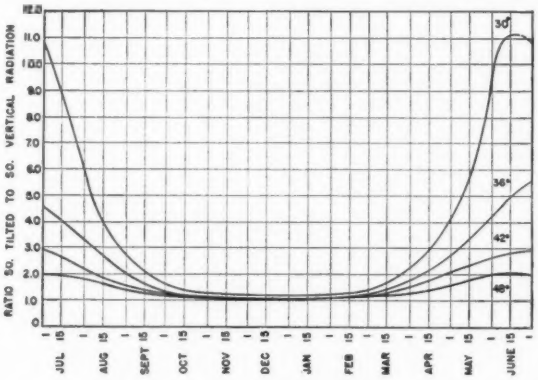


FIG. 5. RATIO OF DAILY TOTAL SOLAR RADIATION OUTSIDE THE ATMOSPHERE INCIDENT UPON A SOUTH-FACING SURFACE TILTED FROM THE HORIZONTAL AT AN ANGLE EQUAL TO THE LATITUDE PLUS 21 DEG TO THAT INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE AT VARIOUS NORTH LATITUDES

by 430/435 for December and February, 425/435 for November and March, and 420/435 for October and April. These corrections allow for the varying distance between the earth and the sun. Hutchinson and Chapman recommend the use of Fig. 6 for practical engineering applications in most parts of the United States. It should be borne in mind, however, that this curve was constructed for a relatively dust-free atmosphere and for sea level, and thus does not apply to industrial areas or mountainous locations.

Sky Radiation during Cloudless Days in Winter: The direct solar radiation incident upon a particularly oriented surface during cloudless days in winter may be estimated by Fig. 6 with modification for the angular effect of the sun's rays. However, this figure does not include sky radiation which results from the absorption by the atmosphere of a portion of the direct solar radiation and the consequent re-radiation of a part of this to the earth. Whereas direct solar radiation is specular, or beamed, sky radiation is of diffuse nature and is generally of much longer wavelength than direct solar radiation.

TABLE 2—DISTRIBUTION OF SOLAR RADIATION, BTU PER (HOUR) (SQUARE FOOT). INCIDENT UPON HORIZONTAL AND SOUTH-FACING VERTICAL SURFACES OUTSIDE THE ATMOSPHERE ON JANUARY 15 AND JULY 15 AT 42 DEG NORTH LATITUDE

SUN TIME	JANUARY 15[a]				JULY 15[b]			
	Altitude β Deg-Min	Wall Solar Azimuth α Deg-Min	$I_H = I_n \sin \beta$	$I_S = I_n \cos \alpha \cos \beta$	Altitude β Deg-Min	Wall Solar Azimuth α Deg-Min	$I_H = I_n \sin \beta$	$I_S = I_n \cos \alpha \cos \beta$
5:00 a.m.	—	—	—	—	—	—	—	—
6:00	—	—	—	—	14 15	—	100	—
7:00	—	—	—	—	25 9	—	172	—
8:00	6 0	54 18	49	251	36 16	87 20	239	13
9:00	14 22	42 54	107	307	47 17	75 48	297	67
10:00	20 59	30 0	155	350	57 37	60 18	341	108
11:00	25 18	15 30	185	377	66 0	36 18	369	133
12:00	26 49	0 0	195	386	69 35	0 0	378	142
1:00 p.m.	25 18	15 30	185	377	66 0	36 18	369	133
2:00	20 59	30 0	155	350	57 37	60 18	341	108
3:00	14 22	42 54	107	307	47 17	75 48	297	67
4:00	6 0	54 18	49	251	36 16	87 20	239	13
5:00	—	—	—	—	25 9	—	172	—
6:00	—	—	—	—	14 15	—	100	—
7:00	—	—	—	—	—	—	—	—

[a] I_n = 433 Btu per (hr) (sq ft).

[b] I_n = 406 Btu per (hr) (sq ft).

The data available in the literature on sky radiation, particularly wintertime radiation, are very meager and, to some extent, inconsistent. The A.S.H.V.E. Research Laboratory at Cleveland has made observations of sky radiation received by variously oriented surfaces during summertime cloudless days.[10] A limited amount of data on sky radiation in summer and winter during cloudless days, based on measurements of the U. S. Weather Bureau at Washington, D. C., has been presented by I. F. Hand.[11] Hand's data, however, are for a horizontal surface only. In Fig. 7 the ratio of direct solar radiation to diffuse sky radiation during typical winter cloudless days is presented as a function of the solar altitude angle. Fig. 7 was constructed from Hand's data.

Practically no data were found in the literature for sky radiation received by a south-facing vertical surface during cloudless days in winter. To estimate

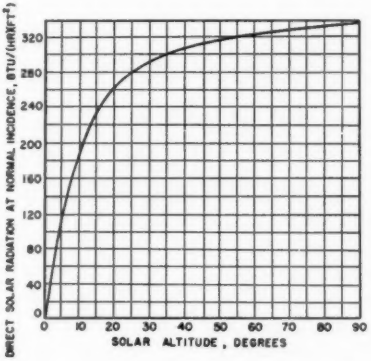


FIG. 6. DIRECT SOLAR RADIATION INCIDENT UPON A SURFACE PERPENDICULAR TO SUN'S RAYS AT SEA LEVEL ON THE EARTH DURING CLOUDLESS DAYS IN JANUARY

this quantity, Fig. 8 was constructed, using Fig. 7 together with the ratio of sky radiation received by a south-facing vertical surface to sky radiation received by a horizontal surface calculated from the summer data given in THE GUIDE. This ratio should be essentially the same for summer and winter. In Fig. 8 the ratio of direct solar radiation at normal incidence to diffuse sky radiation incident upon a south-facing vertical surface is presented as a function of the solar altitude angle.

Daily Total Solar and Sky Radiation Available at the Earth's Surface During Cloudless Days at 42 Deg North Latitude for October 1 to May 1: The calculation of hourly rates of solar and sky radiation incident upon variously oriented surfaces on the earth during cloudless days may be carried out with use of Figs. 6, 7, and 8. The results of such calculations for a horizontal surface and a south-facing vertical surface at 42 deg north latitude are shown in Table 3 together with sample calculations.

A comparison of Table 2 with Table 3 indicates that depletion by the atmosphere is greatest at the lowest altitude angles where the length of path of the sun's rays through the atmosphere is longest. At 8 a.m. and for a horizontal surface, approximately 42 percent of the solar energy available outside the atmosphere is depleted by the atmosphere when the altitude angle is 6 deg; at 12 noon only 19 percent is depleted when the altitude angle is 26 deg, 49 min. The relative depletion for a south-facing vertical surface is even greater than that for a horizontal surface. At 8 a.m., about 67 percent of the available solar radiation is depleted by the atmosphere, and at 12 noon, about 30 percent is depleted.

An important conclusion may also be made regarding the relative amount of radiation which is incident upon an east-facing or west-facing surface during cloudless days. Such a surface is sunlit only a little more than 4 hrs on January 15 at 42 deg north latitude or half the time of illumination of the south-facing

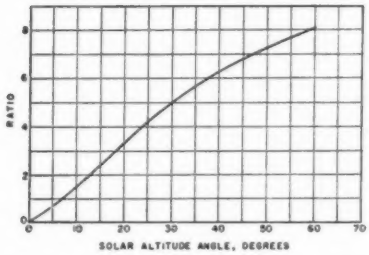


FIG. 7. RATIO OF DIRECT SOLAR RADIATION TO DIFFUSE SKY RADIATION INCIDENT UPON A HORIZONTAL SURFACE DURING TYPICAL WINTER CLOUDLESS DAYS

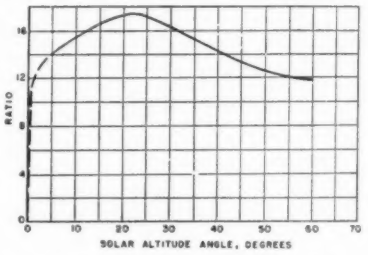


FIG. 8. RATIO OF DIRECT SOLAR RADIATION AT NORMAL INCIDENCE TO DIFFUSE SKY RADIATION INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE DURING TYPICAL WINTER CLOUDLESS DAYS

vertical surface. In addition, for those hours when the angular effect of the sun's rays is most favorable, the relative depletion by the atmosphere is greatest. It is evident that an east-facing or west-facing surface is quite ineffective for collection of winter sunshine.

Once hourly rates of solar and sky radiation incident upon a particular surface during cloudless days have been calculated, the daily totals may be found by application of Simpson's rule. Calculations were carried out at weekly intervals for the period October 1 to May 1 for 42 deg north latitude for a horizontal surface and for a south-facing vertical surface. The daily totals thus determined are shown in Fig. 9 together with the curve for the daily total direct solar radiation incident upon a south-facing vertical surface outside the atmosphere. The curve for daily total solar and sky radiation incident upon a south-facing vertical surface during cloudless days has a rather unusual shape. As December 21 is approached from either side, the angular effect of the sun's rays becomes more favorable for interception of direct solar radiation by a south-facing surface. However, at the same time, depletion by the atmosphere increases, due to the lower altitude angles of the sun. Sky radiation values also decrease as

TABLE 3—DISTRIBUTION OF SOLAR AND SKY RADIATION, BTU PER (HOUR) (SQUARE FOOT). INCIDENT UPON HORIZONTAL AND SOUTH-FACING VERTICAL SURFACES ON THE EARTH DURING CLOUDLESS DAYS ON JANUARY 15 AT 42 DEG NORTH LATITUDE

Sun Time	Altitude β Deg Min	Wall Solar Azimuth Deg-Min	Normal Incidence Radiation I_o	Horizontal Surface[a]			South-facing Vertical Surface[b]		
				Direct Solar Radiation I_D	Sky Radiation I_S	Total Radiation I_T	Direct Solar Radiation I_D	Sky Radiation I_S	Total Radiation I_T
7:00 a.m.	—	—	—	—	—	—	—	—	—
8:00	6 0	54 18	126	13.1	15.4	28.5	73.1	8.8	81.9
9:00	14 22	42 54	232	57.5	24.5	82.0	164.7	14.0	178.7
10:00	20 59	30 0	266	95.2	26.8	122.0	215.2	15.3	230.5
11:00	25 18	15 30	280	119.6	28.2	147.8	244.2	16.3	260.5
12:00	26 49	0 0	286	129.0	28.6	157.6	255.1	16.8	271.9
1:00 p.m.	25 18	15 30	280	119.6	28.2	147.8	244.2	16.3	260.5
2:00	20 59	30 0	266	95.2	26.8	122.0	215.2	15.3	230.5
3:00	14 22	42 54	232	57.5	24.5	82.0	164.7	14.0	178.7
4:00	6 0	54 18	126	13.1	15.4	28.5	73.1	8.8	81.9
5:00	—	—	—	—	—	—	—	—	—

[a] Sample calculations for 12:00 o'clock are shown below:

I_o = 286 Btu per (hr) (sq ft) from Fig. 6.
$I_D = I_o \sin \beta$ = (286) (0.451) = 129 Btu per (hr) (sq ft).
I_D/I_S = 4.5 from Fig. 7.
I_S = 129/4.5 = 28.6 Btu per (hr) (sq ft).
$I_T = I_D + I_S$ = 157.6 Btu per (hr) (sq ft).

[b] Sample calculations for 12:00 o'clock are shown below:

I_o = 286 Btu per (hr) (sq ft) from Fig. 6.
$I_D = I_o \cos \beta \cos \alpha$ = (286) (0.892) (1.0) = 255.1 Btu per (hr) (sq ft).
I_o/I_S = 17 from Fig. 8.
I_S = 286/17 = 16.8 Btu per (hr) (sq ft).
$I_T = I_D + I_S$ = 271.9 Btu per (hr) (sq ft).

December 22 is approached, and these two effects combine to indicate the two peaks for this curve.

SOLAR AND SKY RADIATION AVAILABLE IN UNITED STATES AT EARTH'S SURFACE DURING ALL DAYS

General Factors Affecting Availability of Radiation: Although it is possible to calculate, with considerable accuracy, the total amount of solar radiation available outside the atmosphere and the total amount of solar and sky radiation available at the earth's surface during cloudless days, these calculations are primarily of value in guidance in solar energy studies, and are of little value

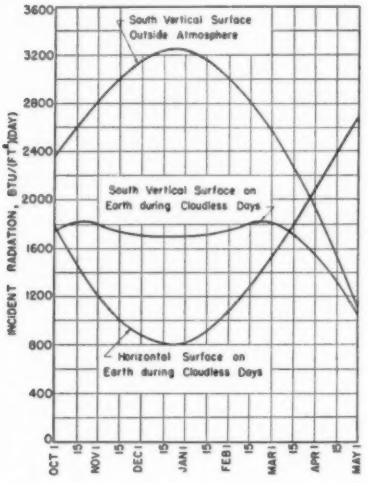


FIG. 9. DAILY TOTAL SOLAR AND SKY RADIATION INCIDENT UPON HORIZONTAL AND SOUTH-FACING VERTICAL SURFACES AT 42 DEG NORTH LATITUDE

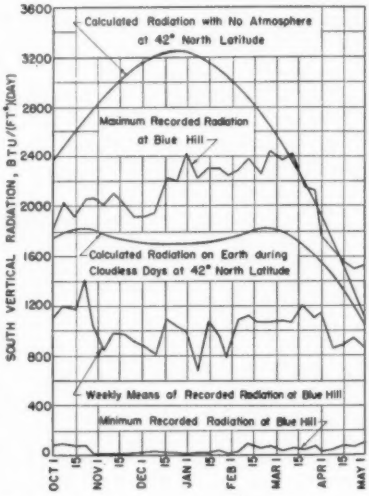


FIG. 10. CALCULATED AND RECORDED DAILY RADIATION INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE

in engineering design of systems for winter collection of solar energy. The effect of clouds on depletion of radiation is probably the greatest single deterrent to the theoretical estimation of actual radiation available. There is no mathematical way to estimate the depletion of radiation by clouds since the amount and type of cloudiness cannot be predicted. Therefore, it is necessary to combine theoretical considerations with actual recorded solar and sky radiation values at different locations in order to predict the engineering availability of solar energy with even exploratory accuracy.

The number of hours of sunshine and percentage of possible hours for the months of December and January are shown in Table 4 for several cities. This table illustrates the widely varying amounts of sunshine in the United States, with December extremes of 45 hr and 16 percent for Erie, Pa., to 254 hr and

82 percent for Yuma, Ariz. *Technical Paper No. 12* of the U. S. Weather Bureau[12] presents these data for approximately 200 cities in the United States. An interesting observation from this publication is that regions immediately south of the Great Lakes, including such cities as Erie and Cleveland, have relatively low amounts of winter sunshine due evidently to cloud formations originating over the Great Lakes. It is also observed that the majority of the stations reported in Washington and Oregon show very low amounts of sunshine in December and January. Thus, these two regions, the Great Lakes and the Pacific Northwest, are relatively poor locations for utilization of winter sunshine. For the country as a whole, excepting these two regions and the southwestern part of the country, about 50 percent of possible sunshine appears normal for December and January.

Atmospheric pollution such as smoke and dust constitute another factor difficult to take into consideration. During a dust storm, incident radiation may be reduced by 90 percent or more.[13] The quantity of dust in the atmosphere over

TABLE 4—AVERAGE NUMBER OF HOURS OF SUNSHINE AND PERCENTAGE OF POSSIBLE SUNSHINE HOURS FOR TYPICAL CITIES DURING DECEMBER AND JANUARY

CITY	DECEMBER		JANUARY	
	Hours	Percent	Hours	Percent
Boston, Mass.	134	48	144	49
Cleveland, Ohio	68	24	86	29
Erie, Penn.	45	16	65	22
Lincoln, Neb.	154	54	171	57
Los Angeles, Calif.	220	72	219	70
Madison, Wis.	101	36	125	43
Miami, Fla.	211	65	213	65
Nashville, Tenn.	127	42	136	43
New Orleans, La.	146	46	161	49
Sault Ste. Marie, Mich.	57	21	77	27
Seattle, Wash.	58	22	71	26
Yuma, Ariz.	254	82	257	81

cities may depend to some extent on the state of economic prosperity. During the depression year of 1932, smoke and dust from manufacturing causes were reduced and there was a marked increase in solar radiation recorded at Weather Bureau stations.[14]

Although limited in extent, actual recorded daily totals of solar and sky radiation as received at various stations of the U. S. Weather Bureau are more revealing and more significant in guiding engineering calculations than are vast amounts of inadequate theoretical calculations. Table 5 lists the U. S. Weather Bureau's network of stations recording such data for a horizontal surface, as well as the year in which the records were begun. The oldest solar station is Madison, Wis., started in 1911. Of the 54 stations listed in Table 5, 29 started their solar observations in 1949 or later.

The Weather Bureau has adopted the Eppley thermoelectric pyrheliometer as its standard recording instrument. Until July of 1941, the weekly means of daily total radiation recorded by the Weather Bureau stations were published in

Monthly Weather Review. From July 1941 through December 1949, the daily totals appeared in the same publication, and since January 1950, the daily totals have been published in *Climatological Data, National Summary.*

Average Solar and Sky Radiation Available on a Horizontal Surface: Table 6 shows the monthly means of daily radiation for December and January obtained by those stations with records of 10 years or longer. The unit indicating the extent of radiation in Table 6 is the langley or one gram-calorie per square centimeter. The entries in Table 6 may be converted to Btu per square foot per day by multiplying them by 3.69.

Fritz and MacDonald[15] have carried out calculations permitting construction of monthly maps showing isolines of average solar and sky radiation received by

TABLE 5—U. S. WEATHER BUREAU STATIONS WHICH RECORD DAILY TOTALS OF SOLAR AND SKY RADIATION RECEIVED BY A HORIZONTAL SURFACE, AND YEAR RECORDS WERE BEGUN

STATION	YEAR	STATION	YEAR	STATION	YEAR
Albuquerque, N. M.	1949	Griffin, Ga.	1950	North Head, Wash.	1949
Apalachicola, Fla.	1949	Hatteras, N. C.	1949	Oak Ridge, Tenn.	1949
Atlanta, Ga.	1949	Indianapolis, Ind.	1949	Phoenix, Ariz.	1949
Blue Hill, Mass.	1933	Inyokern, Calif.	1948	[a]Put-in-Bay, Ohio	1942
Boise, Idaho	1949	Ithaca, N. Y.	1934	Rapid City, S. D.	1949
Boston, Mass.	1944	[a]La Jolla, Calif.	1928	Riverside, Calif.	1933
Brownsville, Tex.	1949	Lake Charles, La.	1949	Salt Lake City, Utah	1946
Charleston, S. C.	1949	Lander, Wyo.	1949	Sault Ste. Marie, Mich.	1950
[a]Chicago, Ill.	1923	Las Vegas, Nev.	1949	Santa Maria, Calif.	1949
Columbia, Mo.	1944	Lincoln, Neb.	1916	Sayville, N. Y.	1949
Columbus, Ohio	1948	Lynn, Mass.	1949	Seabrook, N. J.	1949
Davis, Calif.	1942	Madison, Wis.	1911	Seattle, Wash.	1950
East Lansing, Mich.	1942	Medford, Ore.	1949	St. Cloud, Minn.	1949
East Wareham, Mass.	1942	Miami, Fla.	1949	State College, Pa.	1941
El Paso, Tex.	1949	Nashville, Tenn.	1942	Stillwater, Okla.	1950
Fort Worth, Tex.	1949	[a]New Orleans, La.	1928	[a]Twin Falls, Idaho	1927
Fresno, Calif.	1928	New York, N. Y.	1924	Upton, N. Y.	1950
Grand Lake, Colo.	1948	Newport, R. I.	1937	Washington, D. C.	1914

[a] Publication discontinued.

a horizontal surface in the United States during all days. This was possible since it was determined that a linear relationship existed between the ratio of actual to cloudless day radiation and the percent of possible sunshine. Since the number of hours of sunshine instrumentally recorded was available for a large number of localities based on many years of records, the calculation of actual radiation in those sections where pyrheliometer studies had not been made was permitted.

Average Solar and Sky Radiation Available on a South-facing Vertical Surface at Blue Hill, Mass.: The observatory at Blue Hill, Mass., is the only weather station in the United States which has made actual measurements of solar energy incident upon a south-facing vertical surface. These measurements of the daily total solar and sky radiation plus the radiation reflected from the ground were begun in March, 1945. Fig. 10 shows a graph of the average weekly means of daily totals of solar and sky radiation received by a south-facing vertical surface

TABLE 6—MONTHLY MEANS OF DAILY TOTAL SOLAR AND SKY RADIATION RECEIVED BY A HORIZONTAL SURFACE IN LANGLEYS PER DAY FOR DECEMBER AND JANUARY

STATION	DEC.	JAN.	STATION	DEC.	JAN.
Blue Hill, Mass.	135	153	Madison, Wis.	119	149
Chicago, Ill.	87	106	Nashville, Tenn.	132	138
Davis, Calif.	146	191	New Orleans, La.	200	215
East Lansing, Mich.	88	101	New York, N. Y.	119	129
Fresno, Calif.	158	182	Riverside, Calif.	216	257
Ithaca, N. Y.	103	116	State College, Pa.	120	130
La Jolla, Calif.	228	252	Twin Falls, Idaho	124	166
Lincoln, Neb.	166	187	Washington, D. C.	159	163

at Blue Hill, together with graphs of maximum and minimum recorded radiation at the same location. The curves for calculated radiation with no atmosphere at 42 deg north latitude and for calculated radiation at the earth's surface during cloudless days at 42 deg north latitude are shown on the same graph. The latitude at Blue Hill is 42 deg, 13 min north.

It is interesting to note from Fig. 10 that the maximum curve for Blue Hill is consistently higher than the curve for calculated radiation during cloudless days. This excess may be attributed to the albedo of the ground—the reflection of radiation from the ground cover and the interception of a part of this radiation by the south-facing vertical surface. The sharp increase of the maximum curve during the period December 15 to March 15 is apparently caused by an increase of the albedo by ground snow cover. It is significant that during March and April some of the values of the maximum curve of recorded radiation exceeded those of the calculated curve for no atmosphere.

Fig. 11 shows a series of lines for different weeks of the winter season representing the weekly mean ratio of south-vertical to horizontal radiation at Blue Hill as a function of radiation incident upon a horizontal surface. Although there was some scattering of points, reasonably good correlation was obtained with the linear relationships shown.

Estimated Average Solar and Sky Radiation Available on a South-facing Vertical Surface in the United States: The recorded data at Blue Hill for a south-facing vertical surface may be considered representative of other localities at 42 deg north latitude with reasonably dust-free atmospheres and with about

TABLE 7—DATES AT BLUE HILL, MASS., WHICH HAVE SIMILAR SOLAR ALTITUDE AND AZIMUTH ANGLES AS DECEMBER 15 AND JANUARY 15 AT OTHER NORTH LATITUDES

NORTH LATITUDE DEG	DEC. 15	JAN. 15
30	Nov. 1	Oct. 15
36	Nov. 15	Nov. 8
42	Dec. 15	Jan. 15
48	Dec. 22	Dec. 22

50 percent sunshine during December and January. Because of variation in cloudiness, atmospheric pollution, and latitude, any extension of Blue Hill data to other parts of the country must be recognized as only an approximation.

Of the several methods considered for extension of the Blue Hill data, the one which showed the most promise involved the use of the known distribution of radiation received by a horizontal surface as presented by Fritz and MacDonald together with the south-vertical to horizontal ratio shown in Fig. 11. Table 7 shows dates for Blue Hill which have similar solar altitude and azimuth angles to December 15 and January 15 at various other north latitudes. Although the similarity of angles is not exact in each case, close agreement exists.

Fig. 12 shows the estimated average daily solar and sky radiation incident upon a south-facing vertical surface in the United States during December, and

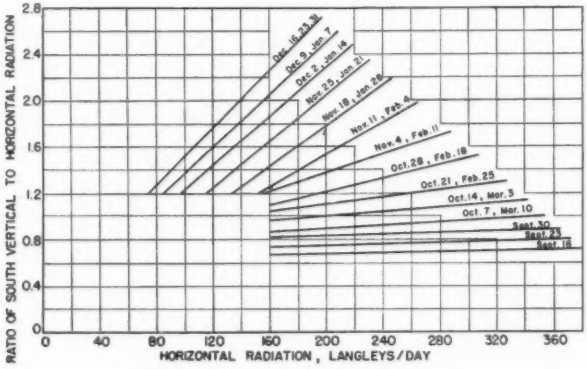


FIG. 11. WEEKLY-MEAN RATIO OF TOTAL RADIATION INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE TO THAT INCIDENT UPON A HORIZONTAL SURFACE AT BLUE HILL, MASS.

Fig. 13 presents a similar map for January. Figs. 12 and 13 were constructed in the following manner: For a given locality, the horizontal radiation was found from the appropriate map of Fritz and MacDonald. For the known latitude of the locality an equivalent date for Blue Hill was obtained from Table 7. Using this equivalent date, a ratio of south-vertical to horizontal radiation was found from Fig. 11. Multiplication of this ratio by the known horizontal radiation gave the estimated incident radiation upon a south-facing vertical surface. This procedure was carried out for approximately 100 localities for both December and January and the isolines of Figs. 12 and 13 were drawn. By this procedure, the cloudiness and atmospheric pollution peculiarities of a particular region are largely accounted for, since the horizontal radiation used was developed from actual recorded data. Variation in the sun's angular position is taken into consideration by the equivalent data determined from Table 7. The albedo of the ground varies somewhat at different locations, particularly where a snow cover is involved, but any error so introduced is of secondary magnitude. Thus, these maps provide reasonably correct data for gross engineering design calculations.

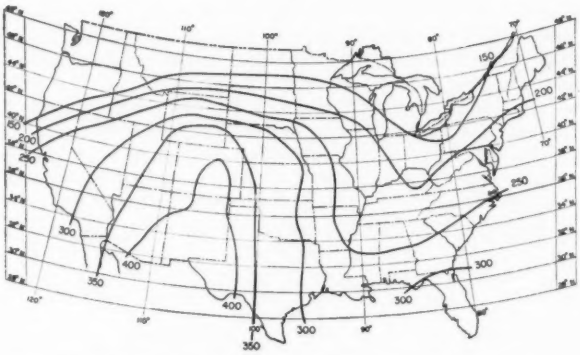


FIG. 12. ESTIMATED AVERAGE SOLAR AND SKY RADIATION IN LANGLEYS PER DAY INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE IN DECEMBER

It is interesting to note from Figs. 12 and 13 that a relatively low amount of radiation is available on a south-facing vertical surface in the regions southeast of the Great Lakes and in the Pacific Northwest. For a given latitude, the available radiation is larger for those states west of the Mississippi River and reaches a maximum in the Rocky Mountains. For both December and January, the maximum radiation appears to be centered in the states of Colorado and New Mexico.

Analysis of Weather Bureau Records for Madison, Wis.; Lincoln, Neb.; Nashville, Tenn.; and New Orleans, La.: Four locations were chosen for study of the comparative design considerations involved in the utilization of solar energy as a partial or total heat source for the heat pump. These locations are

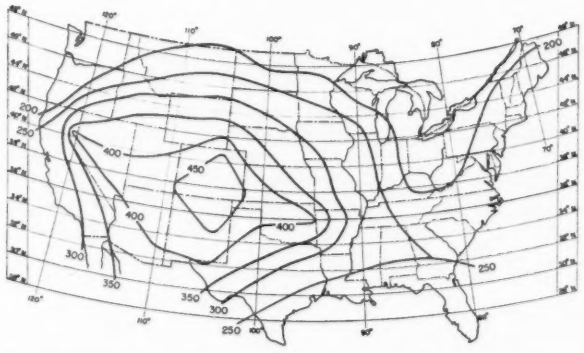


FIG. 13. ESTIMATED AVERAGE SOLAR AND SKY RADIATION IN LANGLEYS PER DAY INCIDENT UPON A SOUTH-FACING VERTICAL SURFACE IN JANUARY

Madison, Lincoln, Nashville, and New Orleans. Although Figs. 12 and 13 present reasonable approximations to the solar and sky radiation available on a south-facing vertical surface for the months of December and January, additional solar statistics are needed for engineering design purposes. For utilization of solar energy for house heating, additional important solar statistics are the possible deviation from the long-time normal radiation for the months of December and January, the sequence of days with above-average and below-average radiation, and the distribution of days in the months of December and January

TABLE 8—PERCENT DEVIATION FROM NORMAL DAILY RADIATION RECEIVED BY A HORIZONTAL SURFACE DURING DECEMBER AND JANUARY AT MADISON, LINCOLN, NASHVILLE AND NEW ORLEANS

YEAR	MADISON		LINCOLN		NASHVILLE		NEW ORLEANS	
	Dec	Jan	Dec	Jan	Dec	Jan	Dec	Jan
1951	−1.0	10.7		18.3		9.3		
1950	28.1	−8.9	6.4	−0.5	0.7	−9.1		
1949	2.8	−16.4	−2.3	−23.9	20.7	−26.3		
1948	18.4	14.9	−6.1	12.4	10.1	26.4		
1947	39.1	3.0	0.4	18.0	11.6	−16.3		
1946	3.0	4.7	3.1	−2.9	12.5	−13.3	13.4	−4.1
1945	10.1	−4.7	−8.3	−23.5	−17.2	−15.6	8.1	
1944	12.1	0.2	−3.9	11.3	−22.7	14.3		16.5
1943	41.5	11.7	18.3	−1.5	−3.5	14.2	5.5	
1942	−14.2	9.0	−32.3	−10.5	−22.7		4.3	34.4
1941	−17.8	−19.1	−19.3	−26.8			10.0	13.6
1940	−22.7	2.7	−29.4	15.4			−9.7	7.0
1939	9.9	−16.1	7.2	1.1			21.4	18.1
1938	−6.5	−22.6	15.1	−3.5			8.1	−1.3
1937	−20.0	−9.9	−1.3	12.7			−5.6	−18.7
1936	16.4	3.7	−10.5	−8.8			−21.4	−1.3
1935	−20.2	−9.5	−25.6	2.0			0.9	0.0
1934	−7.7	−18.1	−7.2	−8.5			6.7	−13.4
1933	−17.0	−12.4	5.7	7.3			6.3	−20.7
1932	−0.4	−33.2	12.7	−5.7			−29.5	−26.7
1931	−23.7	−10.4	−26.6	14.6				

with above-average and below-average radiation. In addition, it is informative to know the distribution of degree-days with various ranges of radiation.

Maria Telkes[16] has carried out some analyses for Blue Hill, Mass. In this section the results of U. S. Weather Bureau records studies for Madison, Lincoln, Nashville, and New Orleans are discussed. The following number of years of records were examined for each of the localities: Madison, 21 years; Lincoln, 21 years; Nashville, 10 years, and New Orleans, 12 years.

Table 8 presents the average monthly percent deviation from daily normal radiation received by a horizontal surface at the four localities in December and January and shows the wide variation in radiation that may be expected. Extremes for Madison are a surplus of 41.5 percent and a deficiency of 33.2 percent, for Lincoln a surplus of 18.3 percent and a deficiency of 32.3 percent, for Nashville a surplus of 26.4 percent and a deficiency of 26.3 percent, and for New Orleans a surplus of 34.4 percent and a deficiency of 29.5 percent.

Table 9 presents the average distribution of days in December and January with less than threshold radiation, with total daily radiation between the threshold value and the monthly average, with total daily radiation between the monthly average and 50 percent greater than monthly average, and with total daily radiation more than 50 percent greater than monthly average. The threshold value was chosen as 40 langleys per day for the south-facing vertical surface for Madison, Lincoln, and Nashville and is assumed to be the daily total radiation below which no useful collection of solar energy would result in a system using a heat pump. No threshold value was selected for New Orleans since it would be a rare day when no solar energy could be collected. Justification for these threshold values will be presented in forthcoming articles concerning the performance of solar collectors. Table 9 shows relatively few days on the average when no radiation could be collected, except in the case of Nashville. As shown,

TABLE 9—AVERAGE DISTRIBUTION OF DAYS IN DECEMBER AND JANUARY WITH VARIOUS RANGES OF RADIATION AS RECEIVED BY A HORIZONTAL SURFACE[a]

CITY	DECEMBER				JANUARY			
	Range A	Range B	Range C	Range D	Range A	Range B	Range C	Range D
Madison	6	9	5	11	5	11	4	11
Lincoln	4	10	6	11	4	9	5	13
Nashville	11	4	3	13	10	6	3	12
New Orleans	—	14	7	10	—	16	8	7

[a] Range A denotes total daily radiation less than threshold value.
Range B denotes total daily radiation between threshold value and monthly average.
Range C denotes total daily radiation between monthly average and 50 percent greater than monthly average.
Range D denotes total daily radiation more than 50 percent greater than monthly average.

Nashville has between 10 and 11 such days on the average and, in this respect, Nashville is an unfavorable locality for winter utilization of solar energy. Table 9 also shows that Nashville has relatively few partly cloudy days and that either very dark days or clear days are usually to be expected.

A study was made to determine for the four localities the sequences of December and January days with less than threshold radiation, with less than monthly average radiation, and with greater than monthly average radiation. This information is necessary in estimating the amount of heat storage required for a house heating system utilizing solar energy. This study indicated that on a basis of 21 years of data, Madison had one eight-day sequence of December days with less than threshold radiation and as many as 23 consecutive days with daily radiation less than the January average; on a basis of 21 years of data, Lincoln had one eight-day sequence of January days with less than threshold radiation and as many as eleven consecutive days with less than the January average; on a basis of 10 years of data, Nashville had one sequence of 11 days with less than threshold radiation; and, on a basis of 12 years of data, New Orleans had one sequence of 16 December days with less than average radiation. This investigation indicated that sequences of very dark days with less than threshold radiation are more frequent at Nashville than at any of the other three cities. Fortunately, except in the case of Nashville, sequences of days with above-

average radiation are more common than sequences with less than average radiation. This is an important advantage in systems designed to utilize solar energy and employing heat storage.

A study was made to determine the distribution of degree-days with various ranges of horizontal radiation at the four localities. Two important conclusions were drawn from this study. Days which have more than the average number of degree-days are usually days with above-average radiation. This is particularly true for very cold days. Furthermore, those days with less than threshold radiation are often days having less than the average number of degree-days.

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Part II—Utilization of Solar Energy for House Heating

THE UTILIZATION of solar energy for the partial heating of homes is not new. For a number of years, the so-called *solar house,* involving large south-facing windows placed to admit a maximum of winter sunshine and with protection from the summer sun by roof overhang, has received fairly common application in modern architecture. Such a design may or may not effect a

decrease in the actual fuel bill. When the sun is shining, fuel will be saved and the house may even overheat. However, at night and during cloudy days a large heat loss will occur through the glass areas and contradictory reports on the effect upon the fuel bill are found in the literature.[1]

Major disadvantages of the solar house are the excessive heat losses through the glass areas, inability to store surplus solar energy available during sunny days for use at night and during cloudy days, and lack of control of the entry of sunshine. These disadvantages may be obviated by use of a roof or wall type collector together with heat storage.

Hottel and Woertz[2] and Hesselschwerdt[3] have given results on two experimental houses built at the Massachusetts Institute of Technology. Each house employed a roof type collector and water heat storage. The house investigated

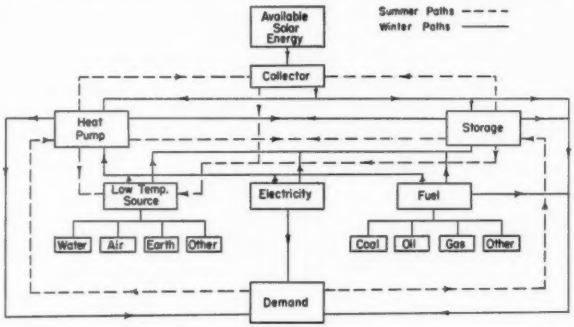


FIG. 1. POSSIBLE PATHS OF ENERGY FLOW FOR YEAR AROUND AIR CONDITIONING SYSTEMS WHICH UTILIZE SOLAR ENERGY AS A WINTER HEAT SOURCE

by Hottel and Woertz included a 17,000 gal water storage tank. Löf[4] has conducted tests at the University of Colorado on a house using a roof type collector and employing crushed rock as the heat storage medium.

Maria Telkes has pioneered in the use of chemical compounds for storage of solar energy, making use of the heat of fusion rather than the specific heat effect. In a series of articles,[5-8] she has discussed problems of solar heating, use of chemical compounds for heat storage, and a report of investigations on an experimental solar house at Dover, Mass. The Dover House was of two-story, shed-roof construction with a 720 sq ft collector occupying the south-facing vertical wall of the second floor. Heat storage was provided by 21 tons of Glauber's salt ($Na_2SO_4 \cdot 10H_2O$) and was capable of supplying the heating requirements of about 10 sunless December days.

Solar energy may be utilized in several ways, depending upon the type of collector, the temperature of collection, and the system of heat storage, if any. Fig. 1 shows possible paths of energy flow for systems which utilize solar energy as either a partial or complete heat source. Both winter and summer paths of energy flow are shown.

[1] Exponent numerals refer to References.

For winter operation, solar energy may be collected by one of several means, and if collection temperature is higher than normal room temperature (about 70 F) this energy may flow directly to the demand or may be stored for subsequent use. If solar energy is collected at a temperature lower than 70 F, it must pass to a heat pump where its temperature level is raised, and the heat pump may then reject heat either to the demand or to storage. Storage of heat energy may occur on either the evaporator or condenser side of the heat pump or both. Solar energy may be used either as a partial or a complete heat source. In general cases, heat from supplementary sources is necessary and in any system supplementary heat from a fuel-fired or electrical source may be used. In systems incorporating a heat pump, low temperature heat sources such as ground water, outdoor air, and the earth may serve as supplementary sources.

Summer paths of energy flow are shown in Fig. 1 as summer cooling may frequently be an integrated part of the system in those systems using a heat

TABLE 1—EFFECTIVE TRANSMISSIVITY—ABSORPTIVITY PRODUCT $(\tau\alpha)_t$ FOR A BLACK COLLECTOR PLATE AND SYSTEM OF GLASS PANES FOR JANUARY 15 AT 42 DEG NORTH LATITUDE

SUN TIME	COSINE OF INCIDENCE ANGLE	$(\tau\alpha)_t$		
		One Glass Pane	Two Glass Panes	Three Glass Panes
8:00 a.m.	0.58	0.814	0.705	0.655
9:00	0.71	0.830	0.750	0.695
10:00	0.81	0.855	0.770	0.720
11:00	0.87	0.855	0.780	0.725
12:00	0.89	0.855	0.785	0.730
1:00 p.m.	0.87	0.855	0.780	0.725
2:00	0.81	0.855	0.770	0.720
3:00	0.71	0.830	0.750	0.695
4:00	0.58	0.814	0.705	0.655

pump. Energy may flow from the demand either to the heat pump or to a low-temperature form of heat storage. The heat pump may reject heat to a low-temperature source such as water or outdoor air, to a high-temperature form of storage, or possibly to the solar collector which could act as a heat sink or dissipator in summer. The low-temperature storage may reject heat to the heat pump or possibly to a low-temperature source such as well water. The high-temperature storage may reject heat to a low-temperature source or to the collector.

The main components of a solar energy-heat pump system are the collector, the heat storage facility, and the heat pump (Fig. 1). In addition, transport systems are required for transferring energy from the collector to the demand via the heat pump and the heat storage facility but design information on air and liquid circulation systems is well established. In the following sections, collectors, heat storage systems, and heat pumps are discussed.

SOLAR ENERGY COLLECTORS

Type, Construction, and Orientation of Collector: The collection of solar energy has been attempted for many purposes and several types of collectors

have been developed. One constructed in Germany produced temperatures higher than 6000 F.[9] Abbot[10] has extensively studied the problem of high-temperature collectors. The inherent disadvantage of high-temperature collectors is their low collection efficiency.

In order to collect solar energy for house heating purposes in an efficient manner, it is desirable to keep the temperature of the collector plate at a relatively low level. For such application the flat-plate type of collector with one or more separated sheets of glass in front of the collector plate is the most common. Maria Telkes and E. Raymond[7] at the Dover House used such a collector made of standards sheets of thin galvanized iron painted black and nailed directly to the back of 4- × 4-in. wood uprights. Two glass panes, 1 in. apart, were mounted in front of the collector plate. A duct, 3⅝ in. deep, was provided between the back of the collector plate and an insulated panel closing off the duct. Heat was absorbed from the collector by circulating air. Similar types of collectors with water pipes soldered to the back of the collector plate have also been used.

The best orientation for a collector is a south-facing vertical position. As discussed in Part I of this paper a slightly greater amount of direct solar radiation impinges on a south-facing collector tilted to its optimum angle. Disad-

COLLECTOR NOMENCLATURE

c = a constant, Btu per (hour) (square foot) (Fahrenheit degree).$^{5/4}$
f = ratio of thermal resistance of outer glass pane to that of an average inner pane, dimensionless.
h_w = convection coefficient due to wind, Btu per (hour) (square foot) (Fahrenheit degree.)
n = number of glass panes, dimensionless.
q_c = rate of heat collection by transport medium, Btu per (hour) (square foot).
q_D = rate of incident direct solar radiation, Btu per (hour) (square foot).
q_d = rate of incident diffuse radiation, Btu per (hour) (square foot).
Q_c = total daily energy collected by transport medium, Btu per (day) (square foot).
Q_t = total daily incident radiation, Btu per (day) (square foot).
Q_{th} = total daily threshold radiation, Btu per (day) (square foot).
t_c = average temperature of collector plate, Fahrenheit.
t_o = outdoor air temperature, Fahrenheit.
T_c = average temperature of collector plate, Fahrenheit absolute.
T_o = outdoor air temperature, Fahrenheit absolute.
U_b = overall heat transfer coefficient from collector plate through backside to outdoor environment, Btu per (hour) (square foot) (Fahrenheit degree).
U_g = overall heat transfer coefficient from collector plate through glass panes to outdoor environment, Btu per (hour) (square foot) (Fahrenheit degree).
α_D = absorptivity of collector plate for direct solar radiation, dimensionless.
α_d = absorptivity of collector plate for diffuse radiation, dimensionless.
ϵ_c = emissivity of collector plate for low temperature radiation, dimensionless.
ϵ_g = emissivity of glass panes for low temperature radiation, dimensionless.
η = daily collector efficiency, percent.
θ = time, hour.
σ = Stefan-Boltzmann constant, (0.173) (10^{-8}), Btu per (hour) (square foot) (Fahrenheit absolute).[4]
τ_D = transmissivity of system of glass panes for direct solar radiation, dimensionless.
τ_d = transmissivity of system of glass panes for diffuse radiation, dimensionless.
$(\tau\alpha)_t$ = effective transmissivity—absorptivity product of system of glass panes and collector plate, dimensionless.
$(\tau\alpha)_{t,m}$ = daily mean value of $(\tau\alpha)_t$ dimensionless.

vantages of a tilted collector compared with a south-facing vertical collector are a greater likelihood of being covered by dust and snow, receipt of less reflected radiation from the ground cover in front of the collector, the interception of far more radiation in summer, and greater difficulty in incorporating it into the house structure architecturally.

Efficiency of South-facing, Vertical, Flat-plate Collectors: The steady state performance of a solar collector which has both sides exposed to the outdoor environment is given by the relation

$$q_c = \tau_D\alpha_D\, q_D + \tau_d\alpha_d\, q_d - (U_g + U_b)\,(t_c - t_o) \quad \ldots\ldots\ldots \quad (1)$$

Substituting

$$(\tau\alpha)_t = \tau_D\alpha_D\, q_D/(q_D + q_d) + \tau_d\alpha_d\, q_d/(q_D + q_d) \quad \ldots\ldots\ldots \quad (2)$$

in Equation 1 gives

$$q_c = (\tau\alpha)_t\,(q_D + q_d) - (U_g + U_b)\,(t_c - t_o) \quad \ldots\ldots\ldots \quad (3)$$

If the collector is an integral part of the building so that no loss of useful heat occurs through the backside, then

$$q_c = (\tau\alpha)_t\,(q_D + q_d) - U_g\,(t_c - t_o) \quad \ldots\ldots\ldots \quad (4)$$

Hottel [11] has presented graphs giving values of $(\tau\alpha)_t$ as a function of the cosine of the angle of incidence for collectors having two panes and three panes of clear plate glass. Values of $(\tau\alpha)_t$ for a collector with one glass pane may be calculated by Equation 2 using the data of Hottel and Woertz.[2] Values of $(\tau\alpha)_t$ for January 15 at 42 deg north latitude are shown in Table 1 for collectors with one, two, and three glass panes.

The overall heat transfer coefficient U_g may be calculated from results given by Hottel and Woertz.[2] To a close degree of accuracy,

$$U_g = \frac{1}{\dfrac{n}{c\sqrt[4]{\dfrac{T_c - T_o}{n+f}}} + \dfrac{1}{h_w}} + \frac{\sigma\,(T_c^4 - T_o^4)}{(T_c - T_o)\left(\dfrac{1}{\epsilon_c} + \dfrac{2n + f - 1}{\epsilon_g} - n\right)} \quad \ldots\ldots \quad (5)$$

For a wind velocity of 10 mph, Hottel and Woertz give values of $h_w = 4.0$ Btu per (hr) (sq ft) (F), $f = 0.36$, and for a vertically placed collector $c = 0.135$. Both ϵ_c and ϵ_g may be taken equal to 0.95. Equation 5 was used to calculate values of U_g for $n = 1$, 2, and 3 panes, for t_c ranging from 20 F to 100 F, and for t_o ranging from 0 F to 60 F. The results are shown in Fig. 2 which indicate that U_g is very closely a function of the collector plate temperature t_c only.

The remaining data necessary for solution of either Equation 3 or 4 are hourly rates of total solar and sky radiation incident upon a south-facing vertical surface. Two cases have been adopted: (1) for a typical cloudless day on January 15 at 42 deg north latitude when the daily total incident radiation is 1780 Btu per sq ft as previously calculated (see Part I) and (2) for a partly cloudy day on January 15 at 42 deg north latitude when the daily total incident radiation is 1000 Btu per sq ft. The assumed rates of radiation are shown in Table 2.

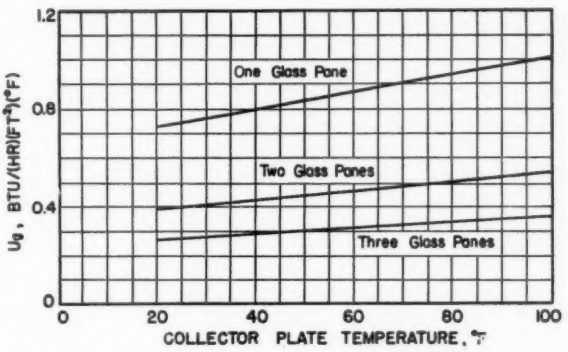


FIG. 2. OVERALL COEFFICIENT OF HEAT TRANSFER U_g FROM COLLECTOR PLATE THROUGH GLASS PANES TO OUTDOOR ENVIRONMENT

The daily collector efficiency in percent is given by the relation

$$\eta = (100)\, Q_c/Q_t \quad \ldots \quad (6)$$

where

$$Q_c = \int_{\theta_1}^{\theta_2} q_c\, d\,\theta \quad \ldots \quad (7)$$

$$Q_t = \int_{\theta_1}^{\theta_2} (q_D + q_d)\, d\theta \quad \ldots \quad (8)$$

Certain approximations are contained in the mathematical developments on collector performance presented here. No shading of the collector face and no dust cover is assumed. The equations are for the steady state and thus no heat storage effects are recognized. Absorption of solar energy by the glass panes is neglected. It is assumed that the outdoor environment to which the

TABLE 2—TYPICAL RATES OF SOLAR AND SKY RADIATION ($q_D + q_d$) IN BTU PER (HR) (SQ FT) FOR A CLEAR DAY AND A PARTLY CLOUDY DAY ON JANUARY 15 AT 42 DEGREES NORTH LATITUDE

SUN TIME	$q_D + q_d$ CLEAR DAY	$q_D + q_d$ PARTLY CLOUDY DAY
8:00 a.m.	82	82
9:00	179	90
10:00	231	115
11:00	261	130
12:00	272	136
1:00 p.m.	261	130
2:00	231	115
3:00	179	90
4:00	82	82

collector radiates is at the outdoor air temperature. All of these approximations, however, should involve but a small error. Hottel and Woertz[2] have shown that for a south-facing collector tilted 30 deg from the vertical the reduction of energy collection due to dust was on the average about one percent. The same authors also found that storage effects were small for the type of collector analyzed here. In regard to daily collector efficiencies, storage effects should partially cancel out. For a good quality of glass, absorption of solar energy is small and the neglecting of it provides conservative results. It may be concluded that collector efficiencies calculated by Equation 6 are representative of maximum efficiencies to be expected from vertical, flat-plate collectors.

Calculations were carried out finding daily collector efficiencies for collectors with one, two, and three glass panes, for various collector plate temperatures,

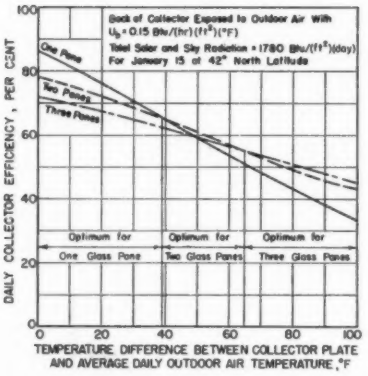


FIG. 3. DAILY EFFICIENCY η FOR A SOLAR COLLECTOR WITH REAR EXPOSED TO OUTDOOR ENVIRONMENT FOR JANUARY 15 AT 42 DEG NORTH LATITUDE ON A CLEAR DAY

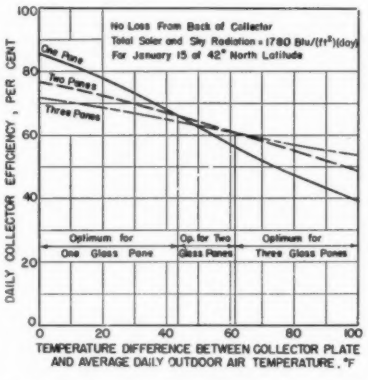


FIG. 4. DAILY EFFICIENCY η FOR A SOLAR COLLECTOR WITH NO LOSS FROM REAR FOR JANUARY 15 AT 42 DEG NORTH LATITUDE ON A CLEAR DAY

for various outdoor air temperatures, for the case of no net loss of useful heat from the back of the collector, and for the case of loss from the backside of the collector with $U_b = 0.15$ Btu per (hr) (sq ft) (F). Hourly values of q_c were found from either Equation 3 or 4 and daily efficiencies by Equation 6. The integrations of Equations 7 and 8 were performed by Simpson's rule. To determine the variation in outdoor air temperature calculations were made for days when the 24 hr average outdoor temperature was 0 F, 20 F, and 40 F. Typical diurnal curves for variation in outdoor temperature were constructed and used for finding t_o in the calculations. The calculations indicated that the daily collector efficiency η was very closely a sole function of the difference in temperature between the collector plate and the daily average outdoor air temperature.

Fig. 3 presents results for daily collector efficiency for the case of $U_b = 0.15$ Btu per (hr) (sq ft) (F) and daily total incident radiation $Q_t = 1780$ Btu per sq ft. Fig. 4 has results for the case of no loss from backside of collector

and $Q_t = 1780$ Btu per sq ft. In Fig. 5 results for the case of $U_b = 0.15$ Btu per (hr) (sq ft) (F) and $Q_t = 1000$ Btu per sq ft are given. Fig. 6 presents results for the case of no loss from backside of collector and $Q_t = 1000$ Btu per sq ft. Each graph shows that there is an optimum range of temperature difference for each system of glass panes which yields the maximum daily efficiency. Comparison of Figs. 5 and 6 with Figs. 3 and 4 shows that when the daily incident radiation is reduced to 1000 Btu per sq ft the daily collector efficiency decreases and the optimum ranges of temperature difference for the different arrangements of glass panes are shifted to the left to smaller temperature differences. Significant are the narrow optimum range for two glass panes, the broad optimum range for three glass panes, and the rapid decrease of collec-

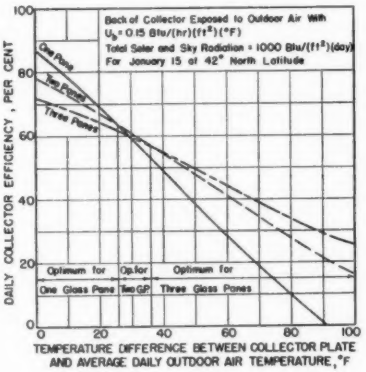


FIG. 5. DAILY EFFICIENCY η FOR A SOLAR COLLECTOR WITH REAR EXPOSED TO OUTDOOR ENVIRONMENT FOR JANUARY 15 AT 42 DEG NORTH LATITUDE ON A PARTLY CLOUDY DAY

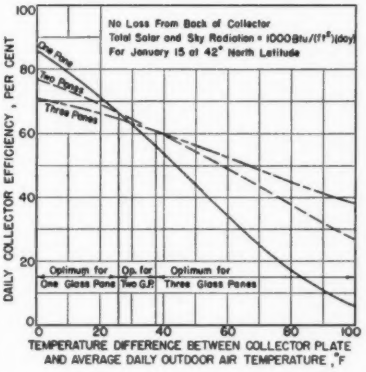


FIG. 6. DAILY EFFICIENCY η FOR A SOLAR COLLECTOR WITH NO LOSS FROM REAR FOR JANUARY 15 AT 42 DEG NORTH LATITUDE ON A PARTLY CLOUDY DAY

tor efficiency with large temperature difference. It is evident that for high collector efficiency a small temperature difference is necessary. It is only in systems using a heat pump that this is possible.

The collector efficiencies presented here have been calculated for January 15 at 42 deg north latitude but the results are applicable to all parts of the United States during December and January because the variation in the transmissivity-absorptivity product $(\tau\alpha)_t$ for a south-facing vertical collector is slight during this period. Figs. 3 and 4 constructed for a daily total radiation of 1780 Btu per sq ft should be applicable for clear days throughout the country.

Threshold Radiation for South-facing, Vertical, Flat-plate Collectors: Valuable information concerning solar collectors includes a knowledge of minimum totals of daily radiation below which no useful collection results. Thus, if Equation 3 is equated to zero

$$q_D + q_d = (U_g + U_b)\,(t_c - t_o)/(\tau\alpha)_t \quad \ldots\ldots\ldots\ldots \quad (9)$$

For those days when no energy is collected considerable cloudiness must occur. Usually there is little variation in outdoor air temperature during daylight hours of a very dark day. Thus, it is reasonable to assume t_o as constant. For a fixed collector plate temperature t_c, the only variable in Equation 9 is $(\tau\alpha)_t$. If a mean value of $(\tau\alpha)_t$ is found, then the daily threshold radiation Q_{th} for the 8 a.m. to 4 p.m. period is given by

$$Q_{th} = 8(U_g + U_b)\,(t_c - t_o)/(\tau\alpha)_{t,m} \quad \ldots\ldots\ldots\ldots \quad (10)$$

Values of $(\tau\alpha)_{t,m}$ were found by the relation

$$(\tau\alpha)_{t,m} = \frac{1}{8}\int_{\theta_1}^{\theta_2} (\tau\alpha)_t\, d\theta \quad \ldots\ldots\ldots\ldots\ldots \quad (11)$$

with the integration performed by Simpson's rule. Values of $(\tau\alpha)_{t,m}$ calculated

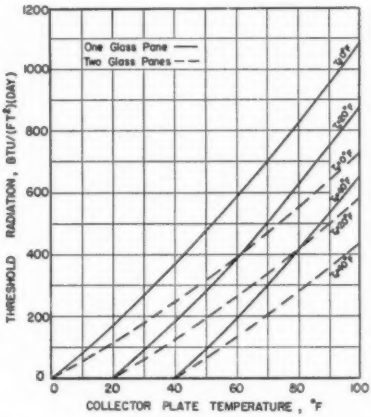


FIG. 7. THRESHOLD RADIATION FOR A COLLECTOR WITH REAR EXPOSED TO OUTDOOR ENVIRONMENT FOR JANUARY 15 AT 42 DEG NORTH LATITUDE

by Equation 11 from the data of Table 1 are 0.845 for one glass pane, 0.762 for two glass panes, and 0.710 for three glass panes.

Fig. 7 shows threshold radiation Q_{th} in Btu per (sq ft) (day) for various collector plate temperatures and various outdoor air temperatures for collectors with one and two glass panes and with the backside of the collector exposed with $U_b = 0.15$ Btu per (hr) (sq ft) (F). As would be expected, the threshold radiation increases rapidly with increase of collector plate temperature or decrease in outdoor air temperature. A disadvantage of a collector with one glass pane is its relatively high threshold radiation and consequent inability to efficiently collect solar energy during partly cloudy days.

Optimum Number of Glass Panes for a South-facing, Vertical, Flat-plate Collector: The optimum temperature of collection depends upon whether a heat

pump is incorporated into the system. In any case, the collector efficiency is highest, and consequently the required collector area is lowest when the collector plate is kept at the lowest possible temperature. If no heat pump is used a collector plate temperature in the order of 100-110 F appears to be minimal for satisfactory results. If a heat pump is included a much lower collector plate temperature may be employed and still provide a relatively high-temperature source of heat when compared to other common heat pump sources. The determination of an optimum collector plate temperature for a system with a heat pump involves consideration of operating costs and the amortized cost of the entire solar energy-heat pump system. Also important is the location and type of heat storage used. If a heat of fusion type of storage medium is used on the evaporator side of the heat pump, the storage temperature should be satisfactory for storage of summer cooling effect also. This requires a storage

TABLE 3—NUMBER OF GLASS PANES FOR MAXIMUM COLLECTOR EFFICIENCY AND THRESHOLD RADIATION FOR VARIOUS NUMBERS OF GLASS PANES FOR A COLLECTOR WHOSE PLATE TEMPERATURE IS 60 F AND BACKSIDE EXPOSED WITH $U_b = 0.15$ BTU PER (HR) (SQ FT) (F)

CITY	JANUARY AVERAGE OUTDOOR AIR TEMP F	$t_c - t_o$ F	NUMBER OF PANES FOR MAX EFF		THRESHOLD RADIATION BTU PER (SQ FT) (DAY)		
			$Q_1 = 1780$	$Q_1 = 1000$	One Pane	Two Panes	Three Panes
Madison	18	42	2	3	405	274	220
Lincoln	25	35	1	2	338	228	184
Nashville	40	20	1	1	193	130	105
New Orleans	55	5	1	1	48	33	26

temperature in the order of 40-60 F. For a generalized typical case where a heat pump is used, a collector plate temperature of about 60 F appears representative.

Assuming a collector plate temperature of 60 F, conclusions may be made concerning the optimum number of glass panes to use in the collector. It is necessary to analyze Figs. 3-7 and to know the average outdoor air temperature in January for the locality concerned. Table 3 shows the number of glass panes which give the maximum collector efficiency for $U_b = 0.15$ for Madison, Wis., Lincoln, Nebr., Nashville, Tenn., and New Orleans, La. Also shown is the threshold radiation for each locality. Considering collector efficiency, threshold radiation, and cost of glass, the optimum number of glass panes appears to be two for Madison and Lincoln and one for Nashville and New Orleans. Generally, two panes are optimum in the northern part of the country and one pane in the southern part when the collector plate temperature is about 60 F.

STORAGE OF SOLAR ENERGY

Requirements for an Ideal Storage Medium: It is evident that if solar energy is to contribute any appreciable part of the heating requirements for a building, energy must be stored during sunny periods for use at night and during cloudy days. Thus, it is necessary to have a quantity of heat storage material suitably placed in the system to accomplish this function.

An ideal heat storage system includes at least the following features:

1. Low cost and ready availability of storage medium.
2. Non-hazardous, non-offensive materials.
3. A storage medium which is non-corrosive to common construction materials.
4. A simple, inexpensive means of packaging the storage medium.
5. High heat capacity per unit volume of storage medium.
6. High heat capacity without large temperature change of storage medium.
7. Heat storage at a moderately low temperature.
8. Low thermal losses from storage medium to ambient surroundings.

Items 1 through 5 and Item 8 are self-explanatory. In Items 6 and 7, high heat capacity without large temperature change and heat storage at a moderately low temperature are required for high collector efficiency and/or high heat pump coefficient of performance.

Specific Heat Type of Storage Materials: In all systems employing a specific heat type of storage material temperature change of the material is necessary

TABLE 4—PROPERTIES OF VARIOUS SPECIFIC HEAT TYPE STORAGE MATERIALS

MATERIAL	SPECIFIC HEAT BTU PER (LB) (F)	DENSITY LB PER CU FT	HEAT CAPACITY BTU PER (CU FT) (F)
Water	1.00	62.4	62.4
Iron	0.11	490	53.9
Concrete	0.27	140	37.8
Brick	0.20	140	28.0

and consequently widely varying temperatures of the medium result. For a given volume and temperature change the heat capacity is proportional to the product of the specific heat and density. Table 4 shows specific heat, density, and heat capacity in Btu per (cu ft) (F) for four low cost materials. It is seen that water is superior to the other materials from a standpoint of heat capacity.

When specific heat materials are analyzed with respect to satisfaction of the features of the ideal medium then common materials such as those of Table 4 are excellent from a standpoint of low cost, ready availability, safety and non-corrosion. Most of these materials need an inexpensive type of container with the exception of water. The main disadvantages of specific heat materials are that they rate poorly with respect to Items 5 through 8 of the features of the ideal medium. Specific heat materials have low heat capacities per unit volume and thus huge volumes are necessary. Inherently, specific heat materials must undergo considerable temperature change and, at least a part of the time, storage temperatures must be relatively high. Thus, specific heat type storage materials appear unsatisfactory if long-term heat storage is required.

Heat of Fusion Type Storage Materials: Considerable interest has been shown in the use of heat of fusion type materials for storage of heat energy. Dr. Telkes proposed the use of these materials, and a system designed by her using Glauber's salt as the storage medium has been discussed. The melting temperature of Glauber's salt is about 88 F and its heat of fusion about 92 Btu per lb.

Calculations for the temperature range of 80 to 100 F indicate that about 7.3 times as much heat may be stored in Glauber's salt as in an equal volume of water.

In order to determine what chemical compounds are presently available that might have possibilities as heat of fusion storage materials in the temperature range of 80-130 F (condenser side storage) and 40-60 F (evaporator side storage), a survey was made of chemical tables and some 40 compounds were found that had melting temperatures in one of these ranges. Both inorganic and organic materials were investigated. However, many of these materials were hazardous, many prohibitively expensive, and complete data lacking on others. No satisfactory material was found for evaporator side heat storage.

Table 5 shows physical properties and estimated cost data on four inorganic compounds suitable for condenser side heat storage. Of the materials shown, Glauber's salt has the least cost per million Btu.

Comparing the features of heat of fusion materials with those of the ideal storage medium indicates that these materials are excellent with regard to high

TABLE 5—PROPERTIES AND COSTS OF VARIOUS HEAT OF FUSION TYPE STORAGE MATERIALS[a, b]

MATERIAL	CHEMICAL FORMULA	MELTING TEMP F	HEAT OF FUSION BTU PER LB	DENSITY LB PER CU FT	LATENT HEAT CAPACITY BTU PER CU FT	COST	
						$ per Ton	$ per 10^6 Btu
Calcium Chloride	$CaCl_2 \cdot 6\ H_2O$	84.2	73.3	104.7	7,670	10	69
Calcium Nitrate	$Ca(NO_3)_2 \cdot 4\ H_2O$	107.8	61.2	113.3	6,950	50	410
Sodium Phosphate Dibasic	$Na_2HPO_4 \cdot 12\ H_2O$	97.0	120.0	94.8	11,400	37	154
Sodium Sulphate	$Na_2SO_4 \cdot 10\ H_2O$	87.8	92.4	91.0	8,400	12	65

[a] Physical properties taken from *Handbook of Chemistry and Physics*, 33rd. Ed., Chemical Rubber Publishing Co., Cleveland, Ohio, 1951.

[b] Costs calculated from data from *Current Market Quotations, Oil, Paint and Drug Reporter*, Vol. 163, No. 1, January 5, 1953, pp. 9-29.

heat capacity per unit volume, high heat capacity without large temperature change, and heat storage at moderately low temperature. However, the materials must be low cost, non-hazardous, and non-corrosive to be satisfactory. Several satisfactory materials are available for storage of heat on the condenser side of the heat pump but practically none are available for evaporator side storage. Evaporator side heat storage is preferable (see Part III). The solution probably lies in the synthesis of materials providing the proper chemical and physical properties and further research is needed in this direction. For example, one of the esters of isophthalic acid, diethyl isophthalate, has a freezing point of approximately 50 F and is probably non-corrosive and non-toxic, though its latent heat is low.

Because of undercooling and temperature stratification, not all of the theoretical heat capacity may be available in heat of fusion materials. In tests[12] conducted for the *Edison Electric Institute* and the *Association of Edison Illuminating Companies* using sodium phosphate dibasic ($Na_2HPO_4 \cdot 12\ H_2O$) results indicated that during the melting cycle 91 percent of the theoretical heat exchange was available and during the freezing cycle only 72 percent was

available. However, additional research is necessary before definite conclusions can be made. Previous research[13] for these same organizations showed that the heat available depended on the shape of the container and that the best results were obtained with small diameter containers.

It is evident that additional research and development are necessary before heat of fusion materials may be applied most effectively. The advantages of this method are such that it appears to be the most promising for long-term heat storage.

HEAT PUMPS

Application of Heat Pumps to Year-Around Air Conditioning: The use of refrigerating equipment for winter heating as well as summer cooling has received much attention during recent years. The main advantages of the heat pump are that it is a non-combustion form of heating offering the maximum degree of cleanliness and safety and the functions of heating and cooling are

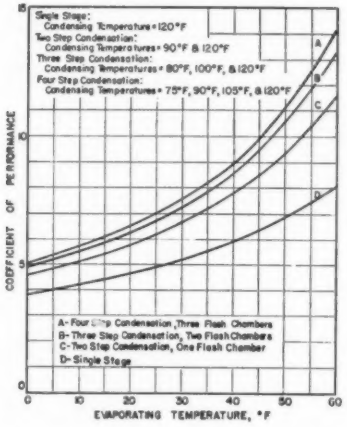


FIG. 8. THEORETICAL HEAT PUMP CYCLE COEFFICIENT OF PERFORMANCE FOR DICHLORODIFLUOROMETHANE (F-12) SYSTEMS USED FOR HEATING A FLUID WITHOUT PHASE CHANGE FROM 60 F TO 120 F (FOR SYSTEMS WITH EQUAL HEATING STEPS IN EACH CONDENSER)

performed by the same system. The principal disadvantage of the heat pump in most parts of the country is its high operating cost for heating since its energy input is power which is generally more costly than fuel. In order for a heat pump to be competitive in operating costs with conventional fuel-fired systems, relatively cheap electricity and a relatively high temperature heat source for the heat pump are required. The use of solar energy as the heat source offers the distinct advantage of a higher temperature heat source than the presently available sources such as outdoor air, ground water, and the earth.

It is of major importance in the design and operation of heat pump systems to realize a high coefficient of performance in order to reduce operating costs. This requires not only a high temperature heat source but also the most advantageous cycle of operation. Practically all heat pump systems in use today employ a single-stage vapor compression cycle of operation. As shown in a previous article,[14] the single-stage vapor compression system is ineffective thermodynamically for heating a fluid through a large temperature range since

the condensing temperature of the refrigerant must be maintained higher than the exit temperature of the fluid heated, and thus a sizeable temperature difference exists in a large portion of the condenser. The use of multiple condensers with flash chambers between allowed the refrigerant to condense in successive stages of lower pressure, thus reducing the mean temperature difference between the refrigerant and the fluid heated and resulting in a marked increase in coefficient of performance. Fig. 8, reproduced from this article, shows the improvement in theoretical coefficient of performance of multiple-step condensation systems compared to the single-stage vapor cycle.

Practical Coefficients of Performance of Heat Pumps: It is advantageous for design purposes to be able to analytically predict the coefficient of performance

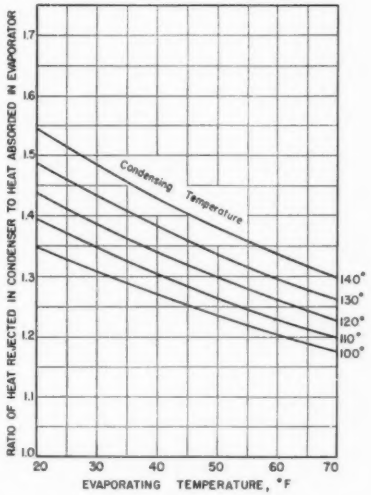


FIG. 9. RATIO OF HEAT REJECTED IN CONDENSER TO HEAT ABSORBED IN EVAPORATOR FOR A SINGLE-STAGE DICHLORODIFLUOROMETHANE (F-12) REFRIGERATION SYSTEM

of actual heat pumps. Values found from an ideal cycle assumption are, of course, too high and it is difficult to know what multiplication factor is needed to give practical values. The following derivation is based upon the use of brake horsepower per ton values taken from manufacturer's rating tables. Two cases will be considered. These are the single-stage vapor compression heat pump and the heat pump arranged for two-step condensation using two condensers, two compressors, one flash chamber, one evaporator and one expansion valve. Heat loss from the compressor and its driving motor will not be included as part of the useful heat rejected.

For the single-stage heat pump

$$\text{COP} = \text{Heat Rejected/Power Required} \quad (12)$$

$$\text{Heat Rejected in Btu per min} = (\text{R})\ (200)\ (\text{Tons}) \quad (13)$$

$$\text{Power Required in Btu per min} = (42.4)\ (\text{BHP})/\eta_m \quad (14)$$

Substitution of Equations 13 and 14 in Equation 12 gives the result

$$\text{COP} = 4.72\ \eta_m\ \text{R}/(\text{BHP/Ton}) \quad (15)$$

For the two-step condensation heat pump

$$\text{COP} = \frac{\text{Heat Rejected in Condenser 1 + Heat Rejected in Condenser 2}}{\text{Power Required for Compressor 1 + Power Required for Compressor 2}} \quad . \quad (16)$$

It can be shown that

$$\text{Heat Rejected in Condenser 1 in Btu per min} = \frac{200\ (\text{Tons})_1\ (R_1)\ (h_e - h_{f,1})}{(h_e - h_{f,2})} \quad . \quad (17)$$

$$\text{Heat Rejected in Condenser 2 in Btu per min} = 200 \left[(\text{Tons})_2\ R_2 + (\text{Tons})_1\ \frac{(h_{f,1} - h_{f,2})}{(h_e - h_{f,2})} \right] \quad . \quad . \quad . \quad . \quad (18)$$

$$\text{Power Required for Compressor 1 in Btu per min} = (42.4)\ (\text{BHP})_1/\eta_{m,1} \quad . \quad . \quad . \quad (19)$$

$$\text{Power Required for Compressor 2 in Btu per min} = (42.4)\ (\text{BHP})_2/\eta_{m,2} \quad . \quad . \quad . \quad (20)$$

If each condenser rejects the same quantity of heat, then

$$\frac{(\text{Tons})_2}{(\text{Tons})_1} = \frac{R_1\ (h_e - h_{f,1}) - (h_{f,1} - h_{f,2})}{R_2\ (h_e - h_{f,2})} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Substitution of Equations 17-20 into Equation 16, assuming that $\eta_{m,1} = \eta_{m,2}$ and making use of Equation 21, gives the result

$$\text{COP} = \frac{9.43\ \eta_m\ R_1\ (h_e - h_{f,1})}{(h_e - h_{f,2}) \left[\left(\frac{\text{BHP}}{\text{Ton}} \right)_1 + \left(\frac{\text{BHP}}{\text{Ton}} \right)_2 \frac{R_1\ (h_e - h_{f,1}) - (h_{f,1} - h_{f,2})}{R_2\ (h_e - h_{f,2})} \right]} \quad . \quad . \quad (22)$$

In Equations 15 and 22 the BHP/Ton values are found from the compressor manufacturer's rating tables at the selected condensing and evaporating temperatures. The ratio R which is the ratio of the heat rejected in the condenser to the heat absorbed in the evaporator is found from Fig. 9 which was constructed using the following assumptions: (1) single-stage system using dichlorodifluoromethane (F-12); (2) 10 F superheat at evaporator outlet; (3) 15 F additional superheating in suction line; (4) 2 psi drop in suction line; (5) 4 psi drop in

HEAT PUMP NOMENCLATURE

BHP = brake horsepower input to compressor, horsepower.
BHP/Ton = ratio of brake horsepower input to tons of refrigeration, horsepower per ton.
$(\text{BHP/Ton})_1$ = BHP/Ton for compressor operated at higher discharge pressure, horsepower per ton.
$(\text{BHP/Ton})_2$ = BHP/Ton for compressor operated at lower discharge pressure, horsepower per ton.
COP = coefficient of performance, dimensionless.
h_e = enthalpy of vapor at evaporator outlet based upon ten degrees of superheat, Btu per pound.
$h_{f,1}$ = enthalpy of saturated liquid at higher condensing temperature, Btu per pound.
$h_{f,2}$ = enthalpy of saturated liquid at lower condensing temperature, Btu per pound.
R = ratio of heat rejected in condenser to heat absorbed in evaporator, dimensionless.
R_1 = value of R for higher condensing temperature, dimensionless.
R_2 = value of R for lower condensing temperature, dimensionless.
η_m = efficiency of electric motor and drive, dimensionless.

compressor suction valves; (6) 15 F additional superheating in cylinder on intake stroke; (7) isentropic compression in cylinder; (8) 6 psi drop in compressor discharge valves; and (9) saturated liquid entering expansion valve.

In both Equations 15 and 22 only the power input to the compressor has been considered. Additional power required for fans, pumps, etc., would have to be included to arrive at a system coefficient of performance.

Fig. 10, calculated by Equations 15 and 22, shows the practical coefficient of performance for single-stage and two-step condensation dichlorodifluoromethane (F-12) heat pumps based upon an assumed motor efficiency of 85 percent. At

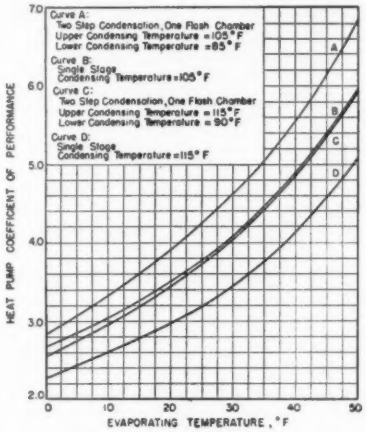


FIG. 10. PRACTICAL COEFFICIENT OF PERFORMANCE FOR DICHLORODIFLUOROMETHANE (F-12) HEAT PUMPS

the higher evaporating temperatures the two-step condensation system has a coefficient of performance in the range of 15 to 20 percent greater than the single-stage system in which the condensing temperature is equal to the upper condensing temperature of the two-step system. Since evaporating temperatures of 50 F or higher are possible with solar energy-heat pump systems, Fig. 10 shows that relatively high coefficients of performance of six or more may be expected.

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Part III—Design and Economics of Solar Energy Heat Pump Systems

A HOUSE designed for utilizing solar energy for heating purposes presents many specialized problems. A large south-facing vertical collector is required for most sections of the country except, perhaps, in the most southerly latitudes, where a south-facing tilted collector is advantageous. In either case, a non-conventional type of construction is dictated. It is desirable to have the collector an integral part of the house structure in regard to construction costs, but separation of the collector from the house causes only a slight reduction in collector efficiency if the rear of the collector is properly insulated and the temperature difference between the collector plate and the outdoor air is moderate.

The design of a solar heated house offers formidable problems to the architect. A large lot is necessary to accommodate a house of such a long, narrow nature; sufficient space in front of the collector is required to prevent shading from neighboring buildings and to prevent reflection from the collector from disturbing inhabitants of nearby buildings. In northerly regions the collector area required may be so large that it dominates the exterior of the house and also complicates the admission of sunlight to the interior. If the collector is to be kept to a moderate horizontal dimension it must be high, or else the vertical dimension may be reduced by allowing the collector to extend beyond the house for some distance. When a south-facing hillside lot is available, a two-level design may be adopted. The upper level may open to the north and the lower level to the south. This scheme offers the possibility of lessening the apparent height of the collector by contrasting it with the hill itself. A house designed for southerly areas is not as great a problem since far less collector area is required. Regardless of the geographical location, any design must be a compromise between a well-engineered solar utilization system and a house whose livable qualities can compete with more conventional types of architecture.

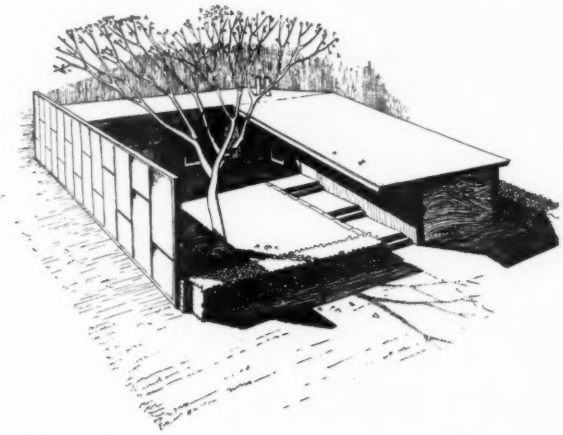


FIG. 1. SOLAR HOUSE WITH SOUTH-FACING VERTICAL COLLECTOR

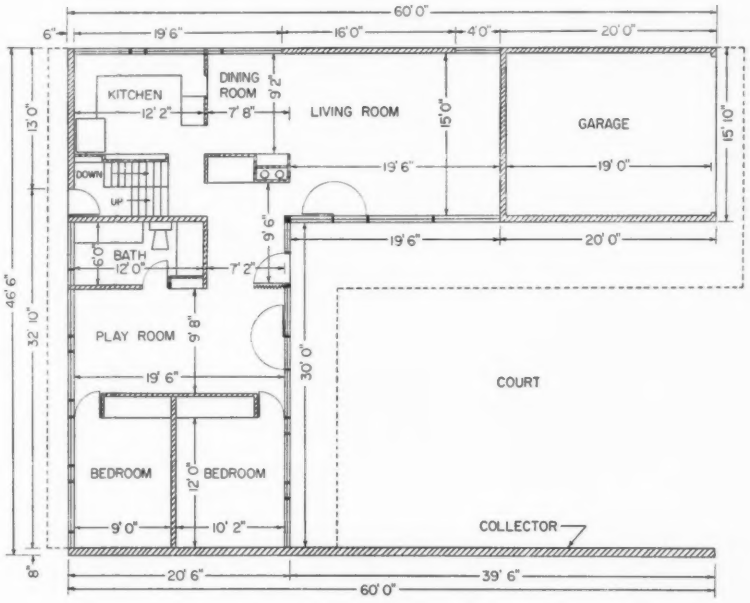


FIG. 2. PLAN OF HOUSE OF FIG. 1

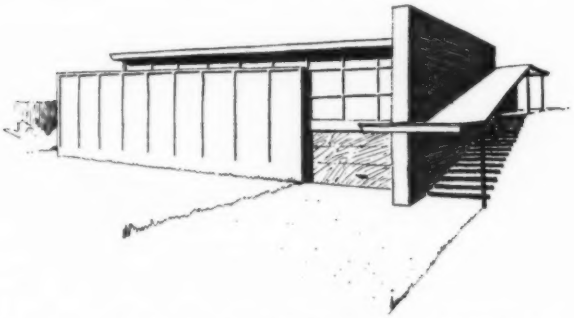


FIG. 3. SOLAR HOUSE WITH SOUTH-FACING VERTICAL COLLECTOR FOR HILL-SIDE LOCATION

Figs. 1, 3, 4, and 5‡ show four different architectural designs of solar houses with flat-plate type, south-facing collectors, and Fig. 2 shows the floor plan of the house of Fig. 1. The design of the house shown in Fig. 1, with the extension of the south-facing vertical collector to form one side of a court, permits full south fenestration of the living room and adequate fenestration on all sides. A full basement is provided for laundry, mechanical equipment, and heat storage facilities. Fig. 3 shows a two-level house using a vertical collector for a south-facing hillside location. The upper level contains the living quarters and the lower level includes a garage and laundry, mechanical equipment, and heat storage facilities. This design results in a collector of large height and consequent shorter length. The designs of Figs. 4 and 5 employ tilted collectors and are primarily for southerly regions of the United States. The design of Fig. 4 is for two levels with bedrooms on the second level and living room and kitchen on the first level. The south side of the lower level has full fenestration. Utility spaces are provided in a basement. The design of Fig. 5 is for three levels with kitchen and dining room on the upper level, living room and entry on an intermediate level, and bedrooms on the lower level.

SYSTEM REQUIREMENTS

Calculation of Collector Area Required: The house shown in Figs. 1 and 2 was used as the example for the calculations. Complete working drawings were

TABLE 1—CALCULATED HEAT LOSSES FOR HOUSE OF FIG. 1

CITY	t_o F	$t_{o,av}$ F	q_d BTU PER HR	q_{av} BTU PER HR
Madison	−15	18	65,000	40,000
Lincoln	−10	25	62,000	35,000
Nashville	0	40	53,000	23,000
New Orleans	20	55	35,000	10,500

‡ These sketches and plans were prepared under the direction of Prof. Harlan McClure of the University of Minnesota School of Architecture.

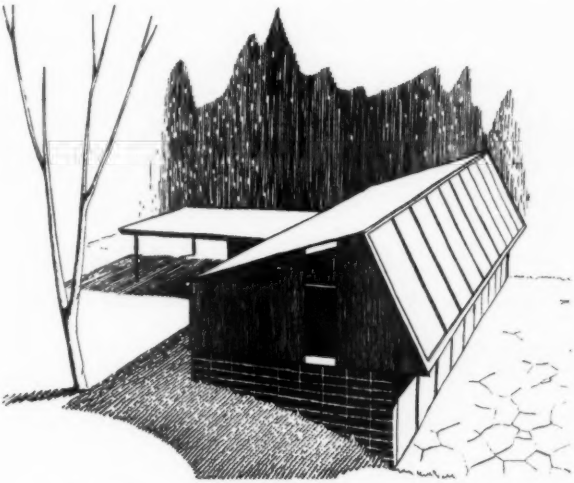


FIG. 4. TWO-LEVEL SOLAR HOUSE WITH SOUTH-FACING TILTED COLLECTOR

available so that heat loss calculations could be made. Insulation was 4 in. of rock wool above the ceiling and 2 in. of blanket insulation in the outside walls. The design heating load was calculated in accordance with Chapter 12 of the HEATING VENTILATING AIR CONDITIONING GUIDE 1953, with infiltration estimated by the crackage method. Table 1 shows the design heat loss and the January average heat loss for Madison, Wis.; Lincoln, Neb.; Nashville, Tenn., and New Orleans, La.



FIG. 5. THREE-LEVEL SOLAR HOUSE WITH SOUTH-FACING TILTED COLLECTOR

TABLE 2—SUMMARY OF CALCULATIONS FOR COLLECTOR AREA

CITY	Q_{av} BTU PER DAY	$Q_{g,av}$ BTU PER DAY	$Q_{s,av}$ BTU PER (DAY) (SQ FT)	η_d	A_c SQ FT
Madison	960,000	79,900	960	0.59	1210
Lincoln	840,000	92,600	1220	0.67	715
Nashville	550,000	40,400	570	0.69	1010
New Orleans	250,000	54,800	810	0.83	226

The January average heat loss was calculated by the equation

$$q_{av} = q_d\,(70 - t_{o,av})/(70 - t_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The temperature difference $(70-t_{o,av})$ was used as an approximation in Equation 1 rather than the usual $(65-t_{o,av})$ because separate credit for solar gains through the window glass was allowed in later calculations.

The collector area is found by a simple energy balance between the total energy which may be collected on an average January day and the January average daily heat loss. Thus

$$A_c = (Q_{av} - Q_{g,av})/Q_{s,av}\,\eta_d R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Table 2 shows the results of calculations for required collector area for Madison, Lincoln, Nashville and New Orleans. The January average daily radiation incident upon a south-facing vertical surface ($Q_{s,av}$) was taken from Weather Bureau data. The collector efficiency η_d shown is based upon a collector plate temperature of 60 F and the January average outdoor air temperature. Two glass panes are assumed for Madison and Lincoln and one glass pane for Nashville and New Orleans. The collector efficiencies and the value of $R = 1.28$ were taken from curves presented in Part II. The value of R was based upon a condensing temperature of 115 F and an evaporating temperature of 50 F. Rational equations were developed for estimating the solar heat gain through the windows at any time of day. Integration of the equations gave values of $Q_{g,av}$.

Table 2 shows that a small collector of 226 sq ft is required for New Orleans, a collector with 715 sq ft of surface is required for Lincoln, and relatively large collectors of 1010 and 1210 sq ft of surface respectively required for Nashville and Madison. The collector shown in Fig. 1 has a gross surface area of 720 sq ft, or approximately suitable for Lincoln.

No suitable correlation exists between collector area and house floor area, or between collector area and house volume, since these ratios vary greatly with architectural design.

TABLE 3—RATIO OF COLLECTOR AREA TO FLOOR AREA AND TO HOUSE VOLUME FOR HOUSE OF FIG. 1

CITY	RATIO OF COLLECTOR AREA TO HOUSE FLOOR AREA	RATIO OF COLLECTOR AREA TO HOUSE VOLUME
Madison	1.0	0.13
Lincoln	0.59	0.074
Nashville	0.83	0.104
New Orleans	0.19	0.023

NOMENCLATURE

A_c = surface area of collector, square feet.
q_{av} = January average rate of heat loss, Btu per hour.
q_d = design rate of heat loss, Btu per hour.
Q_{av} = January average rate of heat loss, Btu per day.
$Q_{g,av}$ = January average rate of solar heat gain through window glass, Btu per day.
$Q_{s,av}$ = January average rate of radiation incident upon a southfacing vertical surface, Btu per day, per square foot.
R = ratio of heat rejected in condenser to heat absorbed in evaporator of heat pump, dimensionless.
t_o = outdoor air design temperature, Fahrenheit.
$t_{o,av}$ = January average outdoor air temperature, Fahrenheit.
η_d = January average daily collector efficiency, dimensionless.

As a matter of interest, these ratios are presented in Table 3 for the example house.

Estimation of Heat Storage Required: In order to estimate the extent of heat storage capacity required, it is necessary to make a study of Weather Bureau records to determine sequence of days below-threshold and below-average incident radiation. Such a study was made for the four localities and was discussed in Part I. From these data, it is possible to estimate the number of days of heat storage needed for the example house of Fig. 1. The approximation used was to consider a day having less than threshold radiation as a full day of heat storage and a day with less than average radiation as one-half day of heat storage.

TABLE 4—ESTIMATED NUMBER OF AVERAGE DAYS OF HEAT STORAGE REQUIRED

SEASON	MADISON		LINCOLN		NASHVILLE		NEW ORLEANS	
	Dec	Jan	Dec	Jan	Dec	Jan	Dec	Jan
1951-1952	3	3		[a]		4		
1950-1951	3	4	[a]	[a]	2	3		
1949-1950	5	3	2	[a]	3	9		
1948-1949	2	4	2	3	5	8		
1947-1948	[a]	6	[a]	[a]	[a]	[a]		
1946-1947	[a]	2	3	[a]	[a]	10		
1945-1946	4	2	2	[a]	6	6	[a]	4
1944-1945	2	6	2	7	12	11		
1943-1944	[a]	3	[a]	3	10	4	5	
1942-1943	4	7	5	6	9	[a]	3	
1941-1942	7	4	4	4			3	[a]
1940-1941	9		9	10			2	2
1939-1940	2	[a]	2	[a]			[a]	5
1938-1939	8	6	[a]	[a]			[a]	[a]
1937-1938	7	8	2	5			3	[a]
1936-1937	6	2	2	3				8
1935-1936	4	6	4	6			[a]	4
1934-1935	5	4	[a]	[a]			[a]	3
1933-1934	2	8	[a]	6			4	5
1932-1933	[a]	3	[a]	[a]			10	10
1931-1932	9	17	6	5				6
Days of Storage Selected		8		7		11		8

[a] Data available, but only overnight storage indicated. Blank places indicate no data available.

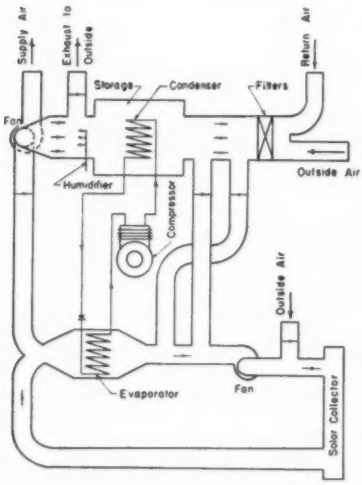


FIG. 6. SCHEMATIC YEAR 'ROUND AIR CONDITIONING SYSTEM USING CONDENSER-SIDE HEAT STORAGE AND SINGLE-STORAGE HEAT PUMP

Table 4 shows the estimated number of average January days of heat storage needed for the four cities and it can be seen that the storage requirements vary widely from year to year with the maximum generally much higher than the average. The number of days of heat storage to be allowed is a matter of judgment. According to Table 4, Madison may need up to 17 days, but there is no assurance that there may not be years which will require more. An average year at Madison requires heat storage for only eight days, and it would be uneconomical to attempt to provide a quantity of heat storage sufficient for any circumstance. An auxiliary heating system must be provided regardless, and therefore the logical solution appears to be to provide storage capacity sufficient for most years and to rely on auxiliary heating for those few unfavorable years when more storage capacity is needed. The number of days of heat storage selected are shown at the bottom of Table 4.

The volume of heat storage medium required for either the condenser side or evaporator side of the heat pump may be estimated if suitable mediums with known heat capacities are available. Several suitable materials are available for condenser-side storage but practically none for evaporator-side storage as dis-

TABLE 5—ESTIMATED HEAT STORAGE REQUIREMENT FOR HOUSE OF FIG. 1

CITY	MILLION BTU	VOLUME CU FT[a]	
		Condenser Side	Evaporator Side
Madison	7.68	768	600
Lincoln	5.88	588	460
Nashville	6.05	605	470
New Orleans	2.00	200	156

[a] Based on heat capacity of 10,000 Btu per cu ft.

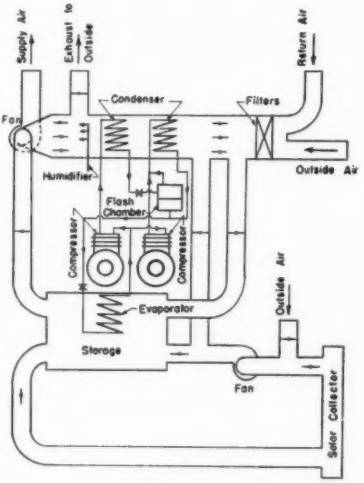


FIG. 7. SCHEMATIC YEAR 'ROUND AIR CONDITIONING SYSTEM USING EVAPORATOR-SIDE HEAT STORAGE AND TWO-STEP CONDENSATION HEAT PUMP

cussed in Part II. There is, however, the definite possibility that such materials may be synthetically compounded. For example such a compound as diethyl isophthalate is probably non-corrosive and non-toxic and has a freezing point of approximately 50 F. However, its latent heat of fusion is too low for satisfactory heat storage use. In the present analysis a hypothetical heat of fusion type storage medium has been chosen for the condenser-side and evaporator-side of the heat pump, each with a heat capacity of 10,000 Btu per cu ft and a suitable melting temperature.

Table 5 shows the estimated volume of storage medium for the four localities. The storage requirement in Btu was found by multiplying the number of January days of storage selected in Table 4 by the January average heat loss in Btu per day from Table 2. The evaporator-side volume was found by dividing the condenser-side volume by $R = 1.28$. It is significant that the storage volumes shown in Table 5 are not unreasonably large and that the largest volume shown is approximately equivalent to a room 10 ft $\times$ 10 ft $\times$ 8 ft high.

Estimation of Heat Pump Requirements: Before the performance requirements of the heat pump may be determined it is necessary to establish the type of heat pump system to be used. In a previous article,[1] it was shown that the single-stage vapor compression system is desirable for the case of constant temperatures of heat source and heat sink, and that the multiple-step condensation system is desirable if the sink temperature is variable.

Fig. 6 shows a schematic year-around air conditioning system using condenser-side storage and a single-stage heat pump. Fig. 7 shows a schematic system using evaporator-side storage and a two-step condensation heat pump. In both systems air is used as the transport medium and the duct dampers are set for winter operation. No auxiliary heat supply is shown but it could be placed on either side of the heat pump, preferably in the storage chamber.

[1] Exponent numerals refer to References.

TABLE 6—HEAT PUMP REQUIREMENTS FOR HOUSE OF FIG. 1 FOR A DICHLORODIFLUOROMETHANE (F-12) SYSTEM USING CONDENSER-SIDE HEAT STORAGE AND SINGLE-STAGE HEAT PUMP

CITY	ENERGY AVAILABLE TO HEAT PUMP		EVAP CAPACITY TONS	COND CAPACITY BTU PER HR	PISTON DISPLACEMENT cfm	BRAKE HORSEPOWER
	Btu per (hr) (sq ft)	Btu per hr				
Madison	217	262,000	21.8	335,000	71	21.8
Lincoln	217	155,000	12.9	198,000	42	12.9
Nashville	236	238,000	19.8	305,000	65	19.8
New Orleans	236	53,000	4.4	68,000	15	4.4

All analyses of this section are premised upon the assumption that the solar energy-heat pump system is capable of utilizing all radiation which can be collected during an average month of January. Thus, a system equipped with condenser-side storage must have a heat pump with a capacity sufficiently large to handle the surplus solar energy available during clear, relatively warm January days. An hourly rate of radiation incident upon a south-facing vertical surface of 275 Btu per (hr) (sq ft) is typical of mid-day radiation during a cloudless January day in most parts of the country. If the day is relatively warm, the collector efficiency approaches the transmissivity—absorptivity product $(\tau\alpha)_t$ of the glass panes and collector plate.

Table 6 shows the heat pump requirements for a system employing condenser-side storage and a single-stage heat pump (Fig. 6). The results are based upon an hourly rate of incident radiation of 275 Btu per (sq ft), $(\tau\alpha)_t$ equal to 0.79 for Madison and Lincoln and 0.86 for Nashville and New Orleans, a condensing temperature of 115 F, and an evaporating temperature of 50 F. The compressor piston displacement and horsepower were found from a typical manufacturer's rating tables.

In a system equipped with evaporator-side storage, the heat pump must have a condenser capacity equal to the design heating load. Table 7 shows heat pump requirements for a system using evaporator-side storage and a single-stage heat pump. Operating conditions are 115 F condensing temperature and 50 F evaporating temperature.

Table 8 shows heat pump requirements for a system using evaporator-side heat storage and two-step condensation heat pump (Fig. 7). Operating condi-

TABLE 7—HEAT PUMP REQUIREMENTS FOR HOUSE OF FIG. 1 FOR A DICHLORODIFLUOROMETHANE (F-12) SYSTEM USING EVAPORATOR-SIDE HEAT STORAGE AND SINGLE-STAGE HEAT PUMP

CITY	COND CAPACITY BTU PER HR	EVAP CAPACITY TONS	PISTON DISPLACEMENT cfm	BRAKE HORSEPOWER
Madison	65,000	4.2	15	4.2
Lincoln	62,000	4.0	14	4.0
Nashville	53,000	3.4	12	3.4
New Orleans	35,000	2.3	8	2.3

TABLE 8—HEAT PUMP REQUIREMENTS FOR HOUSE OF FIG. 1 FOR A DICHLORODIFLUOROMETHANE (F-12) SYSTEM USING EVAPORATOR-SIDE HEAT STORAGE AND TWO-STEP CONDENSATION HEAT PUMP[a]

CITY	CONDENSER CAPACITY BTU PER HR		EVAP CAPACITY TONS	PISTON DISPLACEMENT cfm		BRAKE HORSEPOWER	
	Cond 1	Cond 2		Comp 1	Comp 2	Comp 1	Comp 2
Madison	32,500	32,500	4.4	8.0	6.0	2.4	1.2
Lincoln	31,000	31,000	4.2	7.7	5.7	2.3	1.2
Nashville	26,500	26,500	3.6	6.3	5.1	1.9	1.1
New Orleans	17,500	17,500	2.4	4.3	3.3	1.3	0.7

[a] Condenser 1 and Compressor 1 operate at higher condensing temperature; Condenser 2 and Compressor 2 operate at lower condensing temperature.

tions are 115 F upper condensing temperature, 90 F lower condensing temperature, and 50 F evaporating temperature. Although the ratio R of heat rejected in the condenser to heat absorbed in the evaporator is closely equal to that of the single-stage system, a more accurate value of $R = 1.23$ was used instead of 1.28 used in the calculations for Tables 6 and 7. Again, the compressor displacement and horsepower were taken from manufacturer's rating tables.

Comparison of Table 6 with Tables 7 and 8 shows that a system using condenser-side storage requires equipment about five times larger than a system using evaporator-side storage. Comparison of Table 7 with Table 8 shows that a two-step condensation heat pump requires somewhat smaller displacement and horsepower than a single-stage heat pump.

It appears without question that evaporator-side storage is preferable, and a two-step condensation heat pump is preferable to a single-stage system. Slightly smaller displacement and power capacity are required and a higher coefficient of performance may be realized.

Estimation of Auxiliary Heat Required: The calculation of collector area presented previously was based upon an assumed incident radiation equal to the long-time January average. However, average daily radiation in January varies widely from year to year, and negative deviations from the long-time average of 30 percent or more may be expected. In addition, the estimation of heat storage required was based upon a sufficient quantity for most years but not enough for the worst years on record. For both these reasons auxiliary heating will be required. The auxiliary heating system must have sufficient capacity to provide the entire heat source for several consecutive days. Usually such periods are

TABLE 9—ESTIMATED AUXILIARY HEATING SYSTEM CAPACITY IN BTU PER HOUR FOR HOUSE OF FIG. 1

CITY	CONDENSER SIDE	EVAPORATOR SIDE
Madison	40,000	31,000
Lincoln	35,000	27,000
Nashville	23,000	18,000
New Orleans	10,500	8,200

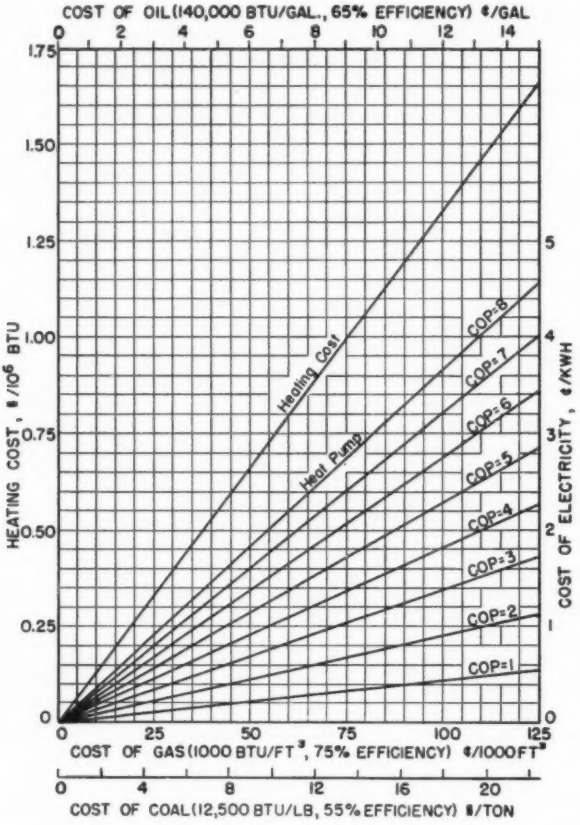


FIG. 8. EQUIVALENT COSTS FOR HEATING WITH THE HEAT PUMP AND GAS, OIL AND COAL-FIRED SYSTEMS AND HEATING COST PER MILLION BTU

days with above average temperature. It appears that a capacity equal to the average January heat loss in Btu per hour is adequate. Table 9 shows auxiliary heating system capacities for both the condenser-side and evaporator-side of the heat pump. It is desirable to supply the auxiliary heat to the storage medium.

COST ANALYSIS

The first cost of a conventional heat pump system is comparable to, or less than, the first cost of other types of year-around air conditioning systems, depending upon the expense of procuring the heat source. It is not possible to make a detailed study of the first-cost of solar energy heat pump systems here.

TABLE 10—RESIDENTIAL COSTS OF ELECTRICITY AND PETROLEUM FUELS AT MADISON, LINCOLN, NASHVILLE, AND NEW ORLEANS

CITY	ELECTRICITY ¢ PER KWH	NATURAL GAS ¢ PER 1000 CU FT	NO. 2 GRADE FUEL OIL ¢ PER GAL
Madison	1.5	90	14.2
Lincoln	1.6	50	11.5
Nashville	0.4	75	13.8
New Orleans	2.0	55	12.0

However, Maria Telkes and E. Raymond[2] have given some cost data on the solar house built at Dover, Mass.

It is of interest to compare the electricity cost for the heat pump with the fuel cost for conventional systems using natural gas, fuel oil and coal. Fig. 8 allows direct comparison of unit electricity cost for the heat pump with unit costs for gas, oil, and coal for delivery of the same quantity of heat at equal cost and provides for graphically determining the heating cost per million Btu.

For example, consider a heat pump with a Coefficient of Performance (COP) of 4 and an electricity cost of 2 cents per KWH. For delivery of the same quantity of heat at equal cost, gas must cost $1.10 per 1000 cu ft, coal about $19.75 a ton, and oil about 13.2 cents per gallon. (In all of these cases it costs $1.46 to deliver one million Btu of heat.)

The economic feasibility of a heat pump in regard to operating cost may be readily determined from Fig. 8 if the actual heat pump coefficient of performance and the unit costs for electricity and fuels are known.

Table 10 shows unit costs of electricity, natural gas, and No. 2 grade fuel oil for Madison, Lincoln, Nashville, and New Orleans. Each city has a residential rate schedule for electricity and natural gas and the costs shown in Table 10 are for those parts of the rate schedules in which the consumption of electricity for the heat pump and the consumption of natural gas for space heating would usually appear.

Table 11 shows the estimated fuel and heat pump costs for an average winter season (October 1 to May 1) for the four localities. The heat pump was taken as a two-step condensation type operated at 115 upper condensing temperature, 90 F lower condensing temperature, and 50 F evaporating temperature. The estimated actual coefficient of performance from Fig. 10 of Part II is 5.9. In

TABLE 11—ESTIMATED HEATING COSTS FOR HOUSE OF FIG. 1 FOR THE HEAT PUMP AND FUEL-FIRED SYSTEMS FOR AN AVERAGE WINTER SEASON AT MADISON, LINCOLN, NASHVILLE, AND NEW ORLEANS

CITY	HEAT REQUIRED 10^6 BTU	COST IN DOLLARS		
		Heat Pump	Gas	Oil
Madison	126.5	94	162	199
Lincoln	106.0	78	72	135
Nashville	64.2	19	65	98
New Orleans	20.2	20	15	27

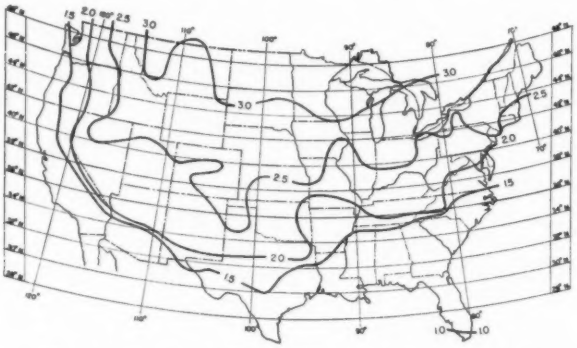


FIG. 9. RELATIVE DESIGN HEATING LOAD FOR THE UNITED STATES

figuring the electricity cost for the heat pump, the first 250 kwhr consumed per month were charged to other electrical appliances and in figuring the cost of natural gas the first 3500 cu ft consumed per month were charged to other gas appliances.

The heat pump costs shown are electricity costs for the compressor only, and no estimates of power consumption for transport of the fluid from the collector to the evaporator are included. Since this additional power consumption would be very small compared with that of the compressor, the values of Table 11 are in correct order of magnitude.

Table 11 shows that for the four localities the heat pump results in lower heating costs than fuel-fired systems, with the exception of natural gas systems at Lincoln and New Orleans. At Nashville and Madison the heat pump has substantially lower operating costs. The reason for the favorable showing of the heat pump is, of course, the relatively high evaporating temperature of 50 F

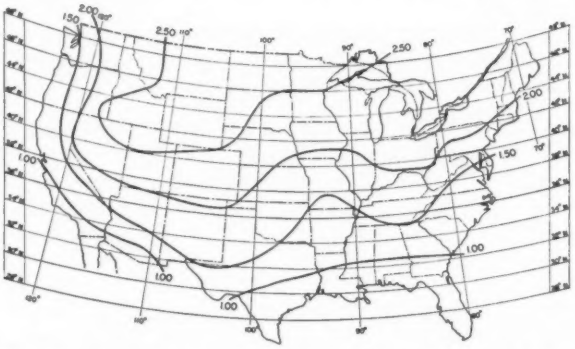


FIG. 10. RATIO OF DESIGN HEATING LOAD TO DESIGN COOLING LOAD FOR THE UNITED STATES

assumed. However, in solar energy-heat pump systems, an evaporating temperature of 50 F may easily be attained.

ADAPTABILITY OF SOLAR ENERGY HEAT PUMP SYSTEMS IN THE UNITED STATES

Any home in the United States may be heated entirely by a solar energy heat pump system if a sufficiently large collector, heat pump, and heat storage facilities are provided. However, it has been shown that at Lincoln, Neb., a locality which has rather severe winters, a house with a design heating load of 62,000 Btu per hr may be heated during average winters with a system equipped with a 700 sq ft collector, a 4 hp heat pump, and a hypothetical but probably attainable heat of fusion type storage material requiring a volume equivalent to a small

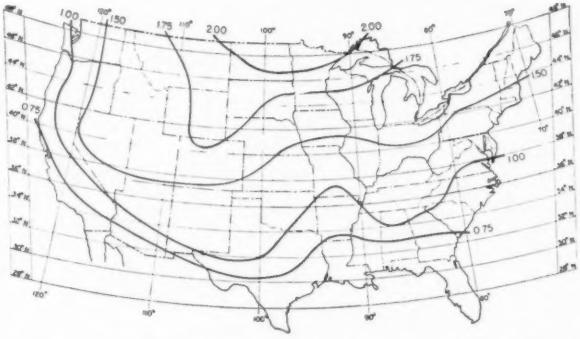


FIG. 11. RATIO OF TONS CAPACITY REQUIRED FOR WINTER HEATING (120 F CONDENSING TEMPERATURE, 40 F EVAPORATING TEMPERATURE) TO TONS CAPACITY REQUIRED FOR SUMMER COOLING FOR A DICHLORODIFLUOROMETHANE (F-12) SYSTEM WITH NO HEAT STORAGE

basement room. An auxiliary heating system of moderate size is required for supplemental energy during the infrequent, very cloudy winters.

It has also been shown that a solar energy heat pump system may result in heating operating costs substantially less than conventional fuel-fired systems. With the accelerated depletion of the world's petroleum resources it is reasonable to expect the relative operating economy of solar energy heat pump systems to become more favorable in the future. It is true at present that such systems would have a high first cost because of the need for custom building of the solar energy system, but standardization of collectors and heat storage units should reduce these costs and provide a system probably competitive in first cost and cheaper in operating cost than fuel-fired systems.

In order to discuss the feasibility of solar energy heat pump systems, it is necessary to analyze the demand for heating and cooling in the United States as well as the available radiation. Heating load and cooling load calculations were carried out for a conventional type house for approximately 100 localities. The same amounts of insulation were used in all calculations.

Fig. 9 shows the relative design heating load for the country. The maximum rate calculated was 3.38 at Duluth, Minn. Fig. 10 shows the ratio of design heating load to design cooling load. With the exception of a narrow belt of southern states and the extreme southwestern section, all regions of the country show a higher design heating load than cooling load. Fig. 11 shows the ratio of tons capacity for the refrigeration system (heat pump) that is required for winter heating to tons capacity required for summer cooling for systems with no heat storage. Winter operating conditions of 120 F condensing temperature and 40 F evaporating temperature for the heat pump were assumed. Fig. 11 was constructed by dividing each of the points calculated for Fig. 10 by a ratio of heat rejected in the condenser to heat absorber in the evaporator of 1.34. Fig. 11 shows that a broad belt of southern states and the Pacific Coast region

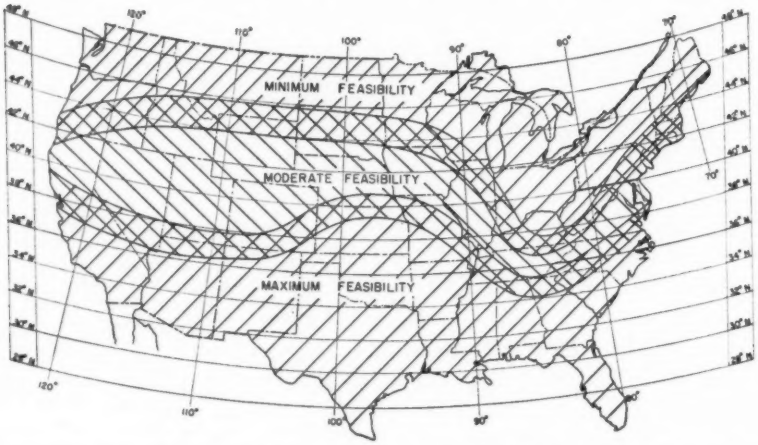


FIG. 12. RELATIVE FEASIBILITY OF SOLAR ENERGY HEAT PUMP SYSTEMS IN THE UNITED STATES

required a greater refrigeration system capacity for summer cooling than winter heating. The New England, Cornbelt, and Middle Rocky Mountain regions require about 50 percent greater capacity for winter heating, and a narrow section of Minnesota and North Dakota require about twice as much capacity for winter heating as for summer cooling.

It is necessary to consider the seasonal demand for heating and cooling as well as the peak demand. The seasonal heating requirement is proportional to the number of degree days for the locality. Degree-day data are well established and are available in tabular form in THE GUIDE[3] and in map form in *Heating and Ventilating's Engineering Databook*.[4]

There is no simple way of correlating the seasonal cooling demand for as yet no completely satisfactory method has been published. According to *Heating and Ventilating's Engineering Databook*, the seasonal cooling requirement of some residences has been found to be proportional to the degree hours above 85 F and a map showing these data is given.

The feasibility of solar energy heat pump systems in the United States may now be discussed by considering the data presented here on heating and cooling demand and data on the availability of solar radiation presented in Part I. For maximum feasibility a locality should have a ratio of tons capacity for winter operation to tons capacity for summer operation of unity or less; a large amount of winter radiation incident upon a south-facing surface; a moderate number of degree days in winter; a large number of degree hours above 85 F in summer; and relatively cheap electricity.

Fig. 12 shows three zones of relative feasibility for the United States. The boundaries between the zones are approximate, since a considerable degree of judgment was exercised in their assignment. Fig. 12 was constructed by considering availability of radiation, the winter heating demand, and the summer cooling demand.

The dip of the zone of minimum feasibility to the south and east of the Great Lakes is due to the low amount of solar radiation in winter resulting from excessive cloudiness. For the same reason the zone of minimum feasibility was extended to include the Pacific Northwest. The peak of the zone of maximum feasibility in the Midwest is due principally to the high number of summer degree hours and the relatively large amount of solar radiation available in winter.

CONCLUSIONS AND RECOMMENDATIONS

The conclusions and recommendations which follow are drawn from the entire paper.

Because of the steady depletion of our fossil fuel reserves, new sources of energy must be utilized in the foreseeable future to prevent degeneration of our standards of living. The use of solar energy for space heating is realizable and the sun's supply of energy is virtually inexhaustible.

The desirable orientation for a solar energy collector appears to be a south-facing vertical position. In regard to the availability of solar radiation incident upon a south-facing vertical surface in the United States during December and January, the maximum radiation available appears to be in the southwestern section of the country centered in the states of Colorado and New Mexico. A reasonably uniform radiation exists in the Mississippi Valley Region. The Pacific Northwest and the regions immediately south and east of the Great Lakes are characterized by low amounts of winter sunshine because of excessive cloudiness. Of the total daily radiation incident upon a south-facing vertical surface outside the atmosphere during December and January at 42 deg north latitude, about 45 percent is depleted by the atmosphere during cloudless days. Recorded totals of daily radiation at Blue Hill, Mass., indicate that about 70 percent of the radiation incident upon a south-facing vertical surface outside the atmosphere is depleted by the atmosphere during all days. Weather Bureau records show that the total radiation incident upon a horizontal surface at a given locality varies widely from year to year and that negative deviations from the long-time average of 30 percent or more may be expected. Weather Bureau records also show that at many localities sequences of winter days with above-average radiation are more common than sequences of days with below-average radiation and that very dark winter days when little solar energy may be collected are commonly days with above-average temperature. Very cold days

are usually days with above-average incident radiation. These are important advantages in solar utilization systems.

A rational method has been presented for estimating the daily efficiency of a flat-plate type solar energy collector. The efficiencies calculated by this method are indicative of maximum efficiencies which may be expected in an actual collector. When the difference between the collector plate temperature and the daily average outdoor temperature is 50 F or less, daily efficiencies during January of 60 percent or more may be expected on clear days for a south-facing vertical collector. Solar energy heat pump systems may operate with a much lower collector plate temperature than direct solar utilization systems resulting in substantially higher collector efficiencies and smaller required collector surface. Calculations indicate that a flat-plate collector with a south-facing vertical orientation should be equipped with two glass panes in the northern part of the country and one glass pane in the southern part.

Any solar utilization system must include a heat storage facility if solar energy is to contribute more than a token part of the space heating requirements of a building. Because of their large heat capacity per unit volume and large heat capacity without large temperature change, heat of fusion type storage materials appear superior to specific heat or latent heat of vaporization types. In systems incorporating a heat pump, the heat storage facility should be placed on the evaporator side.

A rational design procedure has been presented for estimating collector area, heat storage volume, heat pump specifications, and auxiliary heating capacity. This procedure may be used for any locality for which the proper solar statistics are available. In many localities a solar energy heat pump system may require components of moderate size. At Lincoln, Neb., a locality having rather severe winters, a house having a design heating load of 62,000 Btu per hour may be heated during average winters with a system equipped with a 700 sq ft collector, a 4-hp heat pump, and a heat storage facility using 460 cu ft of a heat of fusion type material (10,000 Btu per cu ft) placed on the evaporator side of the heat pump.

In many localities solar energy heat pump systems would result in lower heating costs at the present time than conventional fuel-fired systems. Substantially lower heating costs for such systems were calculated for Madison, Wis., and Nashville, Tenn. With steady depletion of the world's petroleum resources, it is reasonable to expect relative operating economy of solar energy heat pump systems to be more favorable in the future. At present, such systems would have a high first cost because of the need for custom building of the solar energy system. However, with standardization of collectors and heat storage units, it would be possible to reduce these costs.

The main problems associated with the development of solar energy heat pump systems involve solar collectors and heat storage systems. No serious architectural problems appear to exist since a house may be designed to include a moderately large south-facing collector and yet retain architectural appeal and livable qualities equal to those of conventional homes. Solar energy heat pump systems appear feasible in a large part of the United States but further research and development is necessary before such systems may be adopted on a large scale.

The following topics appear to be sufficiently important to justify further research:

1. Pyrheliometric measurements of solar and sky radiation incident upon a south-facing vertical surface in all sections of the country to supplement the recorded data now available from the Blue Hill, Mass., Observatory.

2. Experimental studies on heat pumps using multiple-step condensers and flash chambers as to their coefficient of performance, construction, and operation.

3. Further studies on solar collectors to improve their efficiency, determine their proper construction, and evaluate their operating characteristics.

4. Development of heat of fusion materials suitable for evaporator-side storage, determination of their performance characteristics, and studies on packaging and arranging of these materials.

5. Field studies on complete solar energy heat pump systems.

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DISCUSSION

AUSTIN WHILLIER, Cambridge, Mass. (WRITTEN): It is clear, I think, that the authors have assembled much data in a form particularly useful to the solar engineer. Their study of south-vertical solar incidence and the combination of heat pumps and solar collectors represents a pioneering effort in the field of solar energy utilization.

In general, only daily totals of solar incidence on horizontal surfaces are known, whereas in engineering design work *hourly* totals of solar incidence on *tilted* surfaces are needed. In Table 6 of the first paper in which monthly means of horizontal radiation are tabulated for numerous stations, direct comparison of the values is possible only for the same month and latitude, and hence little can be deduced from these values. There is needed a new series of solar radiation measurements on a national scale which in addition to horizontal and south-vertical solar incidence, would also measure the incidence upon a *fixed* surface tilted southward from the horizontal at an angle equal to the latitude. These data would permit direct comparison of localities of *all latitudes* for the same month. Differences between incidence totals at various localities would then be due only to cloudiness and atmospheric turbidity and to a lesser extent to the length of the theoretical day. The effect of the incidence-angle of the radiation on the surface would be eliminated. Furthermore the calculation of solar incidence upon surfaces tilted at other angles would then be subject to far less error than when horizontal insolation data are used, as is necessary at the present time.

In connection with the tilting of south-facing surfaces, a simple manipulation of equations 1, 2, and 3 of the first paper permits calculation of the angle of tilt from south-vertical that would result in the interception of the greatest amount of radiation. This optimum value of ϕ turns out to be about 1 deg greater than the noontime solar altitude. For the special case of no atmosphere (zero diffuse sky-radiation) the incidence on a surface tilted at the optimum angle is

$$(I_{ST})_{max} = I_S \sqrt{1 + (I_H/I_S)^2}$$

(nomenclature as in Part I). The magnitude of the radical term, in January at 42 deg north latitude, varies from 1.06 at 9:00 a.m. to 1.12 at noon. An analogous analysis for cloudless days will show that in January at 42 deg north latitude a solar collector tilted at the optimum angle ($\phi_{opt} \cong \beta_{noon}$) will intercept about 10 percent more radiation than a south-vertical collector, and will have a 15 to 20 percent greater output. This is a significant improvement in performance.

The conclusion reached by the authors, namely that the improvement in performance of an optimum-tilt collector compared to one on a south-vertical surface is not suffi-

cient to justify use of the tilted collector, is contrary to my experience. Such a decision could be justified only if the cost of a south-vertical collector were less than about 85 percent of the cost of a tilted collector (to account for the 15 percent poorer performance), and provided that sufficient south-vertical wall area were available to satisfy the heating load. In general, at latitudes greater than about 40 deg there is in any structure an acute shortage of usable wall-area for collectors, and the 15 percent additional output of a tilted collector becomes significant. (At lower latitudes the superiority of the tilted collector over the south-vertical collector becomes even more pronounced.)

In discussing Fig. 10 of Part I the authors attribute the difference between the calculated and the maximum-recorded solar incidence values mainly to reflected radiation. While reflected radiation is undoubtedly a contributing factor, part of the difference is due to the fact that Fig. 6, upon which the calculated values are based, represents the *average* of cloudless days rather than the clearest of all cloudless days. To establish the relative importance of these two effects, a set of curves similar to Fig. 10, but a horizontal surface (in which case there is no reflected radiation), should be plotted for comparison with horizontal Blue Hill, Mass., data. Incidentally, my experience indicates that the curve of Fig. 6 recommended by Hutchinson and Chapman is 5 to 10 percent low in non-industrial areas, of which Blue Hill is typical.

Figs. 12 and 13 of the first paper showing isolines of south-vertical solar incidence are a valuable addition to the tools available to the solar engineer. Until measurements of south-vertical incidence are actually made these curves will fill what has heretofore been a considerable gap in our knowledge.

In the discussion of the approximations contained in the equation of collector performance (Part II), it is said that absorption of solar energy by the glass panes is neglected. A little further on it is stated that this leads to slightly conservative results. I may have misunderstood what is meant by this, but the effective transmissivity-absorptivity product of Hottel *does* include the loss due to this absorption of solar energy by the glass, and further, it makes an allowance for the fact that this absorbed energy in the glass is not a total loss to the system since it makes up part of the outward heat loss through the glass.

Regarding the optimum number of glass panes (Figs. 3, 4, 5, and 6 of the second paper) the authors appear to have omitted the possibility of a fourth pane reducing the range of the three-pane optimum. Calculations for 4, 5, and 6 panes would indicate that each has a narrow range.

In Table 3 and subsequent analyses optima have been specified on the basis of thermal performance only, without consideration of costs. This is unfortunate because the inclusion of cost factors results in a significant shift of all optima. For example, with the cost of the additional glass taken into account, the change from one to two panes of glass is not justifiable until the difference between the collector temperature and the average outdoor air temperature is around 50 or 60 F. On this basis then, the economic-optimum number of glass panes for all locations (at the temperatures of operation considered in the papers) would be one.

In regard to the storage problem presented in Part III, no mention is made of the spaces required for passage of the transport-air about the chemical containers in specifying the volume of storage. These spaces may add from 40 to 60 percent to the volume, even in well-packed systems. Also, the allowance of only 10 deg between the collector temperature (60 F) and the storage temperature (50 F) seems a little optimistic. I believe that a study that included both the initial equipment cost and the pumping cost would indicate a difference of between 30 and 50 deg as being desirable. This, again, would radically change the picture because the solar collector would have to operate at a higher temperature.

The assumption of a 10,000 Btu per cu ft hypothetical phase-change storage material is, I think, a little optimistic. The value for ice is only 9,000 Btu per cu ft, and ice has one of the highest known heats of fusion. The heat of fusion for sodium phosphate

dibasic that is given in Table 5 of the second paper is a little misleading since 120 Btu is not the energy exchange at 97 F. Only 85 Btu are involved at 97 F, the remaining 35 Btu being the heat of solution between 97 F and 112 F. It should perhaps be pointed out that the sodium phosphate will perform satisfactorily only in a mechanically-stirred system because of its incongruent melting point.

The comparison of numerous heating systems in Part III is based upon thermal performance and annual operating cost, with no consideration of the influence of initial cost of the installations. I should like to ask if it is the intention of the authors to extend their study to include consideration of the latter costs. While it might well be argued that initial costs may be taken into account at a subsequent stage, they are of such magnitude and importance that the determination of optima should be on an economic basis that includes the combined influence of thermal performance, initial cost and annual operating cost. The conclusions arrived at regarding the feasibility of solar heat pump systems must therefore be interpreted as being subject to no limitation of initial cost, since the analysis is based upon thermal performance only. The figures given in Table 11 of Part III serve only to compare the annual running costs of thermal-optimum and not economic-optimum solar heat pump systems with conventional fuel-burning systems.

R. W. MCKINLEY, Pittsburgh, Pa. (WRITTEN): The author's purpose is clear and important. The information they are presenting is interesting and useful. It is more immediately useful, I believe, than one might assume from a simple consideration of its potential value in *Solar House* design and similar developmental problems related to the depletion of our fossil fuel reserves.

I hope that I can interest the authors in adding similar data for the spring, fall, and summer months and for North, East and West elevations. The year-round data will be of immediate value in current heating, ventilating and air conditioning design. Because of their interest in and familiarity with the Weather Bureau records, I would like to ask that they analyze and comment also on Weather Bureau wind records and their bearing on natural ventilation design. I feel that in many areas, unwarranted reliance is placed by designers on natural ventilation as a means of dissipating unwanted Solar Heat Gain during mild and hot weather. This is becoming increasingly apparent in schools and offices with large areas of side-wall windows and skylights. The authors could help to improve this situation by making a simplified summary of related design conditions readily available.

Because electric lighting systems introduce useful quantities of thermal energy and are needed most at night and on the very dark, overcast days when solar energy is not available, it would be interesting to consider the heating capacity of typical electric lighting systems designed to produce 10, 20, 40 and 80 foot-candle illumination levels. Similarly, in mild weather it is possible that a photoelectric control in the electric lighting system would justify its cost by insuring that the added cooling load of the electric lighting system would not be imposed on a cooling system on the bright, clear days when the fenestration system is sure to provide adequate daylight illumination. A very interesting study along this line is being made by Torrence MacDonald of the U. S. Weather Bureau who has determined that their total solar energy data in Langleys can be converted to daylight foot-candles by applying a multiplying factor approximately equal to 7000.

M. L. GHAI,* West Hartford, Conn. (WRITTEN): The lack of data on the availability of solar energy is a serious handicap in estimating the economics and engineering feasibility of the utilization of solar energy for various purposes. It is of special importance in heating of buildings and other applications where the energy is required at comparatively low temperature level. These applications are presumed to be nearly competitive for several localities, although, it is difficult to estimate precisely their

* Director of Research, Bush Manufacturing Co.

economics due to lack of information required for engineering design. The authors have made an admirable approach toward analyzing the scanty information available.

The paper seems to indicate a very interesting and fortunate fact. Throughout the United States, January is generally considered the worst month as far as heating is concerned. That the month of January is almost always colder than December is indicated by the table of degree days given in THE GUIDE for about 230 stations in the United States, Canada and Newfoundland; the degree days for December are never more than the degree days for January. Table 6 indicates that in all cases presented in the table, the poorest month for the availability of solar energy is December, not January. The same trend is indicated by Table 4 giving average sunshine hours. If this trend can be considered generally true, it is fortunate that the availability of the solar energy is comparatively favorable in the month of January when the heating load is maximum. Apparently the cold weather generally lags behind the minima of the received solar energy.

Table A compares the degree days and the solar energy received for the stations of Table 6 for which the number of degree days were available in THE GUIDE. For all cases, the trend for degree days and the solar energy received in January is more

TABLE A—COMPARISON OF AVERAGE MONTHLY DEGREE DAYS, AND AVERAGE MONTHLY TOTAL SOLAR RADIATION RECEIVED BY A HORIZONTAL SURFACE IN LANGLEYS PER MONTH FOR DECEMBER AND JANUARY

STATION	DECEMBER			JANUARY		
	Total Solar Radiation	Degree Days	Solar Radiation per Degree Day	Total Solar Radiation	Degree Days	Solar Radiation per Degree Day
Chicago, Ill.	2697	1116	2.4	4743	1218	3.9
Fresno, Calif.	4898	562	8.7	6642	573	11.6
Ithaca, N. Y.	3193	1129	2.8	3596	1236	3.9
Lincoln, Neb.	5146	1144	4.5	5797	1242	4.7
Madison, Wisc.	3689	1296	2.9	4619	1451	3.2
Nashville, Tenn.	4092	748	5.5	4278	788	5.4
N. Orleans, La.	6200	304	20.4	6665	323	20.7
New York, N. Y.	3689	940	3.9	3999	1028	3.9
Washington, D. C.	4929	872	5.7	5053	928	5.4

than for December. The ratio of average monthly solar radiation to degree days is generally less for December than for January, although, in some cases it is almost the same. December would thus appear to be a more critical month for heating with solar energy even though the heat load is maximum in January.

Another point of particular importance is the observation of the authors stated in the last paragraph of the paper that very cold days normally have above-average radiation, and the days with less than threshold radiation are often days having less than the average heating requirement. This would tend to even out the variations in the daily supply and demand of the heating energy.

Whereas, Figs. 12 and 13 should prove very useful for engineering designs, a word of caution might be in order. These figures are based on total solar radiation which includes the direct specular radiation, and the indirect diffused or the sky radiation. Most types of solar heat collectors respond very favorably to the direct radiation but their efficiency for collecting sky radiation is very poor. Solar collectors often use plates of ordinary window glass which has excellent properties of trapping solar energy due to its high transmissivity for short wave solar radiations and low transmissivity for the long wave radiations emitted from the inside surfaces of the collec-

tor itself. It thus lets in the solar radiations but does not allow the heat rays to be transmitted outside. Since the diffused radiation is of longer wave length, the glass which is suited for direct solar radiation is poor for the diffused radiation. Furthermore, the transmissivity of glass is best for normal rays only. It is less for other angle of incidence and is very poor for angles of incidence less than the critical angle. The diffused radiations coming at all different angles of incidence are therefore not transmitted properly. Solar collectors which use focussing metallic reflectors, or even straight metallic reflectors, are poorer than collectors using glass. They do not collect any appreciable quantity of diffused sky radiations. It is therefore desirable to treat the direct and diffused radiations separately, or alternatively, as a safe practise, not to rely upon the diffused radiation. However, due to the lack of existing data on diffused radiation, it is doubted if it is possible to draw reliable curves, as in

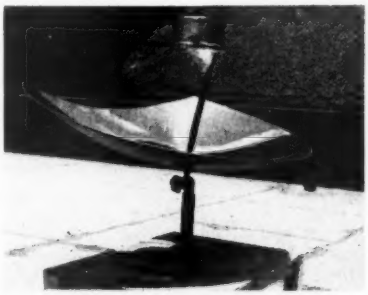


FIG. A. A DEVICE FOR COLLECTING AND UTILIZING SOLAR ENERGY FOR COOKING

Figs. 12 and 13, for the direct and diffused radiation separately, or even to have approximate integrated values of diffused radiations for different localities.

An interesting influence of the diffused radiation was observed during the development of a solar cooker at the National Physical Laboratory at New Delhi, India. It was found that the solar cooker sometimes gave better performance on cold winter days, in spite of larger heat losses from the exposed surface of the cooking utensil, than on warm summer days with a greater amount of total solar radiation. During the summers in New Delhi, the general dust content of the air is greatly increased on account of the frequent wind storms coming from a neighboring desert. The high dust content increases the percentage of diffused radiation even though the solar altitudes are more favorable in summer than in winter. The reflector of the solar cooker did not collect the diffused radiation, and consequently reduced performance was the result. It was found necessary to rely upon the direct solar radiation instead of the total radiation. The development of the solar cooker, shown in Fig. A, has been covered in a series of papers published in the *Journal of Scientific & Industrial Research*, New Delhi, and the commercial production of the cooker has been started in Bombay, India.

With the increasing popularity of year 'round air conditioning in the United States, as indicated by the record sales of air conditioning equipment and still higher sales predicted for the coming years, the study of the use of solar energy for the heat pump is very timely and will probably show a way toward overcoming some of the well known limitations of the heat pump.

It may also be worthwhile to extend the studies to systems combining the use of the heat pump with direct heating with solar energy. For the normal winter weather

conditions, an inexpensive direct solar heating system may suffice, whereas, during the extreme weather conditions, the air conditioning unit run as a heat pump could supplement the heat supply. Such a system would reduce the heavy heat storage requirement which tends to make heating with solar energy alone uneconomical. With the exception of peak load conditions, such a system would still supply heat with no recurring fuel expenses. Considering both the initial capital investment and the recurring expenses, it might give year 'round air conditioning at a most favorable cost.

G. V. PARMELEE, Cleveland, Ohio: I would like to congratulate the authors on having covered a very difficult and complicated subject in excellent and thorough fashion.

In Mr. Ghai's discussion I note the statement that solar heat collectors have a poor performance with respect to diffuse radiation. I assume that this refers to diffuse solar radiation, since glass is opaque to the long-wave diffuse radiation from the atmosphere. Glass, however, is as transparent to the diffuse solar radiation as it is to the direct. However, since diffuse radiation comes from all angles, its average transmittance is less than its transmittance for normal incidence but it is greater for high angles of incidence. For example, two sheets of ordinary glass will transmit about 68 percent of the incident diffuse solar radiation as compared to 76 percent for normal incidence and 66 percent for a direct beam incident at 60 deg. Hence, the authors quite properly take into account the contribution that diffuse solar energy can make. They correctly point out that we need much more investigation as to values of diffuse radiation.

I want to take this opportunity to point out the very great usefulness of the principles of thermal circuitry and circuit analysis*, in analyzing the behavior of such a complex thermal system as that composed of a solar collector, a storage unit and the building served by these units.

A. B. NEWTON, Wichita, Kans.: I was very interested in the paper presented by the authors, since I have been interested in heat pumps for some time. The paper analyzes the savings in our natural resources which would result from solar heat storage as a heat source for the heat pump, and I inferred that it showed the economic practicability of the heat pump with solar storage.

I wonder if anything has been done to carry the analysis a step further and show what the thermal storage does for us in terms of the overall saving of natural resources, bearing in mind, of course, that in the power plant we have a cycle which is the opposite to the heat pump cycle, *i.e.*, we burn fuel to produce electricity.

C. M. ASHLEY, Syracuse, N. Y.: Was the cooling load calculation based upon a 24-hr operation or upon an instantaneous load? Since the present trend is toward 24-hr operation, this is an important point with respect to the size of the system.

The second question is: Where you have heating requirements greater than cooling requirements, would it be possible to deliver a portion of the sunlight gain to the high side of the system, even though at a less efficient rate, in order to make it possible to utilize the solar system without increasing the size of the refrigeration system?

B. P. FISHER, Houston, Texas: I want to ask if money is available to continue this basic research. Its importance to our country with its growing market and consumption cannot be overemphasized. Because we must do more and more with less and less, I think money to further this research must come from industry and from our membership.

R. G. VANDERWEIL, Boston, Mass.: Concerning the actual installation cost of a solar-storage system, I thought it worthwhile to mention a system in one home near Boston, Mass., which was heated for three or four years by use of a solar receiver similar to one of the types discussed. Dr. Maria Telkes, formerly with Massachusetts

* Circuit Analysis Applied to Load Estimating, by H. B. Nottage and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 60, 1954, p. 59).

Institute of Technology, who designed the system, has given information on installation and maintenance cost†.

There was no auxiliary heating system installed at a time when I visited the house for one entire afternoon and evening. My only complaint was that the house was too warm!

AUTHORS' CLOSURE: We are deeply appreciative of the many comments which this paper has prompted. The remarks of Dr. Whillier are particularly interesting and we agree with many of his suggestions. However, although it is important and desirable information, we feel that he has placed overemphasis on the need for hourly totals of solar incidence on tilted surfaces, and that information on incidence on vertical surfaces is of equal importance.

In the third and fourth paragraphs of his discussion, he provides percentage figures indicating the greater output which can be obtained from tilted collectors over south-vertical collectors. However, it should be pointed out that these figures are correct only for comparatively high temperatures of collection and, therefore, are not applicable generally. The output of the collector depends upon the temperature level of collection. The percentage losses are much lower at low temperature collection levels such as assumed in the current articles. In addition, he has given no consideration to the effect of ground cover reflection for which there is still comparatively little available information. This factor favors vertical collectors over tilted collectors. In southern latitudes there will probably be an economic advantage in the use of tilted collectors but we do not feel that there is sufficient evidence that this is true in northern latitudes and particularly at low temperature collection levels. Moreover, if the advantage is only slight, the incorporation of vertical collectors into an architecturally acceptable structure is likely to justify economically the use of vertical absorbers even at the expense of slightly larger collectors.

We do not believe that it is justifiable at this time to include any economic considerations as to the cost of the actual solar system itself, since any attempt to do so will almost certainly be pessimistically erroneous. For example, Dr. Whillier suggests that if the cost of additional glass is taken into account, the change from one to two panes of glass is not justifiable except at temperature differences between the collector and the average outdoor air temperature of about 50 or 60 deg. This, however, does not consider the possibility of the use of a low-cost plastic inner pane. It is our feeling that manufacturing and fabricating costs should not bias technical considerations during a pioneering period.

Dr. Whillier suggests that the allowance of a 10 deg differential between collectors and storage is optimistic. We feel that the 30 to 50 deg which he suggests is pessimistic.

Because of an error in Equation 2 of Part I the ratio of I_{ST} to I_S as shown in Fig. 5 was increased. This increase is slight during winter months but appreciable during the summer. Fig. 5 has been recalculated and the corrected graph is shown. During the December-January period the average ratio is about 1.16 at 30 deg and about 1.04 at 42 deg north latitude. The ratio increases during spring and summer, reaching a maximum of about 11 at 30 deg and about 2.9 at 42 deg.

Although important in some applications, this correction does not change the authors' conclusions concerning collectors used for the purposes intended in these articles. If the collector plate temperature is held relatively low such as 60 F then little advantage accrues from tilting the collector from the south-facing vertical position except in the most southerly latitudes of the country. However, if the collector plate temperature is kept relatively high, such as 100 to 150 F, then a definite advantage results from tilting the collector. At present we are making an experimental study of the performance of flat-plate collectors and one of the variables under investigation will be the effect of tilting the collector.

† See References 5 to 9 of Part II.

The comments of Mr. McKinley are quite pertinent. More comprehensive studies of Weather Bureau records should be made. In addition a detailed consideration of the energy output of electric lighting systems is important both in heating and cooling systems.

Mr. Ghai presents the question of whether December or January is the most critical month for heating with solar energy. It is true as Mr. Ghai points out, that in most localities January is the coldest month and December has the least sunshine incident upon a horizontal surface. However, it is more important to consider the amount of radiation incident upon a south-facing vertical surface and in this respect December compares more favorably with January. In any event, the design of a solar heating system should involve a careful study of both December and January records for the locality. Mr. Ghai's comment about direct and diffuse radiation has been already discussed by Mr. Parmelee. Mr. Ghai's comment about combining the heat pump with direct solar heating is well presented. Such an arrangement may be practicable in many localities.

We wish to thank Mr. Parmelee for his comments, especially those relative to Mr. Ghai's discussion. We concur with Mr. Parmelee that the principles of thermal circuitry could well be adapted to the study of solar heating systems.

In regard to Mr. Ashley's first question, the cooling loads calculated were instantaneous values. Answering his second question, it is possible to deliver a portion of the solar energy to high side storage without increasing the size of the refrigeration system needed for summer cooling. However, such an arrangement would require heat storage on both sides of the heat pump and also a larger solar collector.

Mr. Newton is quite correct that an analysis would be interesting and should be made to show the savings in our natural resources which would result from utilization of solar energy in this manner. However, this was not the intent of the current articles, which were concerned primarily with the feasibility of such systems.

Mr. Fisher of Houston asks if money is to be made available through the Society for continuation of such research. I believe Prof. B. H. Jennings has already answered this question, and the authors are pleased to add that the Society has advised us that a grant has been approved for continuation of our researches.


1503

No. 1503

EFFECTS OF AIR CONDITIONING ON PLANT GROWTH†

By F. W. WENT*, PASADENA, CALIF.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Earhart Plant Research Laboratory of the California Institute of Technology

ONE OF THE most neglected aspects of the cultivation of plants is the control of their environment. Horticulturalists carefully select plants for high productivity and disease resistance; control pests and diseases; use elaborate machinery to prepare the soil, to plant and to harvest; fertilize and irrigate whenever accurate determinations and field experiments indicate the need: but nothing is done about the weather. Yet, of all factors influencing plant performance and crop yield, light intensity and duration, temperature and wind are of the greatest importance.

GREENHOUSES AND THEIR DEVELOPMENT

The only method employed extensively to modify plant environment is the use of greenhouses. Logically speaking, it would be expected that modification of the greenhouse atmosphere would be accomplished by air conditioning, but for several reasons this is not the case: (1) greenhouses are much older than air conditioning; for centuries they were operated without its benefits and the common human trait of conservatism has delayed the general application of air conditioning to greenhouses; (2) air conditioning has been developed and has been considered mainly with human and industrial needs in mind; and (3) a few simple but important modifications to the general practices of air conditioning have to be made before it is applicable to the growing of plants.

Basically greenhouses were developed in cold climates like those of England and the Netherlands, to make it possible to grow plants, such as orchids, native to warmer climates. Later they were used to produce summer flowers and fruits out of season. This meant compliance with the following principles: (1) exclusion of frost during winter; (2) exposure to sufficient light to allow photosynthesis; (3) heating of the air whenever the outside temperature is too low for normal growth; and (4) ventilation to allow entrance of CO_2 and to

† This research work has been carried on under the guidance of the A.S.H.V.E. Technical Advisory Committee on Plant and Animal Husbandry: A. J. Hess, *Chairman;* A. B. Algren, H. A. Borthwick, Samuel Brody, John Everetts, Jr., F. G. Gustafson, S. R. Lewis, H. A. Lockhart, H. H. Mitchell, K. V. Thimann, W. G. Whaley, R. B. Withrow, H. E. Ziel.

* Professor of Plant Physiology, California Institute of Technology.

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cool the greenhouse whenever too much heat from the sun's radiation is trapped under the glass cover.

In the so-called conservatories in use in the 17th century, which were nothing but large rooms with French windows, the heating requirement was satisfied by simple convection of the heat from a stove, placed near the plants. This sufficed to keep them frost-free, but with the development of glass-covered greenhouses considerably more heat and a more efficient means of heat distribution were required. This was first done with long smoke-channels, but in the last 100 years hot water or steam pipes, judiciously distributed through the greenhouse, solved the heating problem.

For cooling, ventilation windows in the eaves of the greenhouse roof were installed, and in some cases acceptable ventilation was achieved by adding side ventilators. The operation of the ventilators is purely manual, and although they do not permit precise control over the greenhouse temperature, they can restrict the rise in temperature at midday. Many growers, however, are loath to open the side ventilators, as insects and infections may enter along with the cooling air.

With the development of air conditioning it was obvious that this could provide the means of controlling the greenhouse atmosphere within the limits required by plants for best performance. The possibilities of air conditioning widen the usefulness of greenhouses tremendously. The usual heating systems provide only for raising the temperature of the houses above that of their surroundings. Conventional greenhouses therefore, are properly called hothouses, and are useful only during cool and cold weather, and then only for plants requiring temperatures higher than those outside. In many parts of the United States, summer greenhouse temperatures may rise to values surpassing the tolerance of many plants, and thus ordinary unventilated greenhouses are effective only approximately 60 percent of the year. This suggests that any means by which greenhouse temperatures can be kept below the tolerance limits of greenhouse crops during the summer will increase the usefulness of greenhouses in the United States.

Before discussing the specifications for air conditioning systems of greenhouses, the exact conditions which are to be maintained should be established, *i.e.*, the optimal growing conditions for plants should be determined. To do this, previously existing knowledge cannot be used as there is little published information concerning the investigation of plant growth under reasonably controlled conditions. For such investigations it is essential to have completely air conditioned greenhouses, and to utilize artificial light. Only by maintaining a particular set of conditions for a sufficient period of time can the response of plants to the conditions be established. Short term experiments are inadequate in this respect, as the work of Blaauw[1] at the Agricultural College in Holland has proved. He found that in addition to the immediate effects of temperature on the development and flowering of tulips, hyacinths and other bulbs, there were delayed effects, which did not show themselves until many months later. Also, effects of light can become visible only weeks or months after exposure.

When, for example, strawberry plants are subjected to two weeks of short days and long nights, the first signs of increased flowering will appear two months

[1] Thermoperiodicity, by F. W. Went (reprinted from *Vernalization and Photoperiodism*—a Symposium, Vol. 1, Edited by Frans Verdoorn, The Chronica Botanica Co., Waltham, Mass., 1948).

later. Therefore, it is almost impossible to deduce the optimal growing conditions for plants from their behavior in the field. For instance, it cannot be said that the high sugar content of sugar beets is due to the high temperatures in July, or the rains in June, or the strong light in August, or the cold nights in September: each of these factors may have had an immediate or a delayed effect,

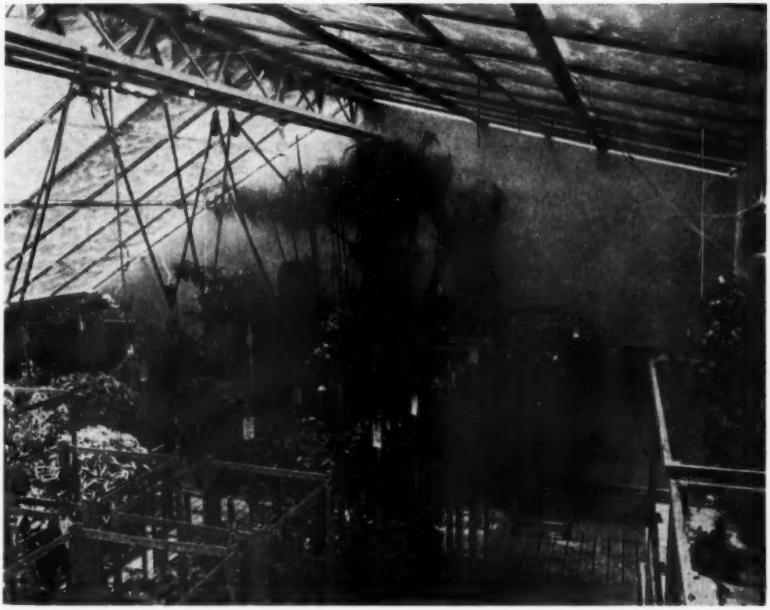


FIG. 1. INTERIOR VIEW OF ONE OF THE ROOMS SHOWING AIR INLET SLOTS IN FLOOR, EXHAUST REGISTERS ON WALL AND WATER FLOWING OVER THE GLASS ROOF. STRAWBERRY PLANTS ARE MOUNTED ON WHEELED TRUCKS. TUBES SUSPENDED FROM CENTER CARRY NUTRIENT SOLUTION AND DEIONIZED WATER

which can be unravelled only by patient research wherein each of these factors is controlled and applied at specified times.

CONTROL OF CLIMATOLOGICAL FACTORS

Before specific recommendations can be made for the construction of commercial air conditioned greenhouses, a set of such greenhouses must be available, in which all climatological factors can be controlled, and in which each factor can be treated as a variable. This was acquired for the first time in the Earhart Plant Research Laboratory (see Fig. 1). With its 20 separate air conditioners a total of 46 different conditions of temperature, humidity, wind, gas content of the air and light are maintained simultaneously. Actually, at any

one time plants are subjected to several hundred different climates, since they are being moved between the 46 conditions, either for a short time or regularly. Thus, any of the 10 different day temperatures can be combined with nine different night temperatures, either to give the large day-to-night fluctuations of dry regions or the small daily temperature range of oceanic climates, or even to reverse the normal conditions of higher day than night temperatures. In this way it was discovered that African Violets (Saintpaulia) grow best at a day temperature of 55 F and a night temperature of 75 F. This behavior, however, is an exception; all other plants investigated thus far do best when the day temperature exceeds that of the night.

For purposes of interpretation of results, most experiments on the effects of temperature fluctuations have been carried out with a constant temperature during the light period and another constant temperature during the dark period. This is also theoretically justified, since in light and darkness, different processes, each with their own optimal temperature, predominate. In special experiments it was found that when plants are subjected to several different temperatures during either the dark or the light period, the plant behaves as if it had been grown at the arithmetical mean temperature, provided the difference in temperature did not exceed 10 deg. With bigger temperature fluctuations growth was less than expected on the basis of the average temperature. The problem of finding the optimal growing conditions for any plant is further complicated by the fact that each plant has its own optimal day and night temperatures and light requirement. Therefore, each species and variety of plant has to be investigated individually. As research progresses, however, certain similarities can be noted, *e.g.,* tomatoes, potatoes and chili peppers respond mostly to variations in night temperature, while peas and strawberries are particularly sensitive to day temperatures. On the other hand, the development of peaches and false hellebore depends entirely on a sufficiently long cold period.

The most extensive work on a particular plant in the Earhart Laboratory has been carried out with tomatoes. They are one of the most important greenhouse crops, and therefore our results have direct practical applicability. From a physiological standpoint the tomato is also a desirable plant, because it does not store food and thus responds within one of two days to a change in climate. Furthermore the tomato is very sensitive to climate, and the big differences in yield from year to year are largely attributable to differences in climate.

EFFECTS OF INDIVIDUAL CLIMATIC FACTORS

Relative Humidity: First a study was made of the effects of the individual climatic factors on the growth and fruiting of the tomato plant. Relative humidity of the air is practically without effect; at 25 percent relative humidity, growth is normal and identical with growth at 80 percent RH (tested at 80 F dry bulb temperature). This is true only when the plants have developed a large root system and are watered regularly.

Wind Velocities: At wind velocities of 5 mph (440 fpm) at plant level, growth is rather strongly inhibited; above 10 miles per hour severe damage to the plants occurs and they die within a few days. Plants kept at wind velocities of 250 fpm at low (250 ft-c) light intensities are sturdier than those grown in relatively calm air at the same light intensity, but it is impossible to say whether this is an effect of air movement or temperature. However, at lower velocities,

such as occur in the air conditioned greenhouses (20-50 fpm), growth is optimal. At still lower velocities (10 fpm or less) growth is reduced again. This is at least partly due to starvation, when the amount of CO_2 within diffusion distance from the stomata becomes limiting. This is a rather important point, for it indicates the degree of air movement necessary in a greenhouse. On dark days lower velocities may be permissible, but any tomato greenhouse should be devised so as to give, during daytime, a minimal rate of air flow past the leaves of 20 fpm. When the greenhouse remains tightly closed (*e.g.*, in freezing weather) an internal source of CO_2 should be provided. If burners could be devised which would burn fuel so completely that no toxic partial oxidation products escape into the air, open burners in the air stream of the supply duct would be ideal, because of the 100 percent heat transfer and the CO_2 supply. In the absence of such open burners the CO_2 supply has to come from the organic matter (manure, bean straw, or corn husks) which is slowly decomposed on or in the soil.

Light Intensity: Light intensity is of great importance in the growth of the tomato plant. A tomato leaf is saturated with light at approximately 1000 ft-c. This value fluctuates somewhat according to the rate at which the plant uses its photosynthates in top, root, and fruit growth, and to the temperature. At low temperatures, leaves are light-saturated at a lower light intensity and therefore greenhouse temperatures should be kept lower on dark days. In non-air conditioned greenhouses this is automatically the case; in air conditioned houses provisions for such temperature adjustments according to light intensity have to be made.

Since a mature plant has leaves over a long stretch of stem, they shade each other, and to have the lower leaves receive a sufficient amount of light, the light intensity in the greenhouse should be several times the saturating intensity for the individual leaf. In general it seems that light saturation for the whole plant occurs only at nearly full sunlight. It is very likely that by losing its older leaves the tomato plant automatically adjusts itself to the prevailing light intensity, but production will always be lower with fewer leaves. When grown in a light intensity of 1500 ft-c., practically no fruit production occurs in an 8-hr daily photoperiod. With 16 hours of light per day, optimal fruit production occurs in this low light intensity at a day temperature of 55 F and a night temperature of 60 F (for the Ailsa Craig variety). In full daylight the optimal day temperature is 73 F and the optimal night temperature 60 F.

Whereas the length of daily illumination is of importance when the light intensity is low, it becomes of less and less consequence at higher light intensities. Therefore in summer, growth and fruit production in the tomato is not limited by light, but in winter it definitely is. Experiments are now under way to find to what extent sugar applications can substitute for this light requirement during winter.

INFLUENCE OF DAY AND NIGHT TEMPERATURES

Night Temperatures: The most important environmental factor in the growth and fruiting of tomato plants is the night (or dark) temperature. Unless this is kept for a sufficiently long time (about 8 hr) nightly for 5-10 days within a rather narrow range, none or few fruits develop, irrespective of all other con-

ditions. This range varies somewhat with varieties and changes with the age of the plants, but as a typical example the behavior of the Ailsa Craig variety will be described. Closely resembling this variety in behavior are Michigan State, Tuckqueen and Essex Wonder. These are all greenhouse varieties, which will produce fruit at light intensities which would be insufficient for field varieties. They also have a slightly lower optimal night temperature than field varieties like Stone, Pearson, Rutgers, etc.

Germination and Transplanting Temperatures: The seeds of tomatoes germinate best at 80 F. When the seedlings are above ground, their fastest growth during the first week is still at nycto-temperatures around 80 F, and then in the course of the next 2 or 3 months, this optimal night temperature decreases gradually to 60 F. This is one, and perhaps the most important, reason why

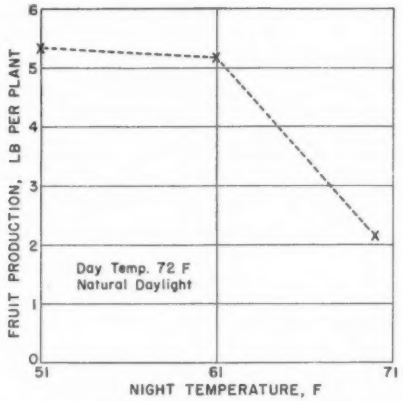


FIG. 2. RELATIONSHIP BETWEEN NIGHT TEMPERATURE AND TOTAL FRUIT PRODUCTION OF THE AILSA CRAIG TOMATO

tomatoes are always transplanted. A greenhouse would have to be kept very warm for the first month to get good seedling growth. A small section, when kept for seedlings only, can be kept at a higher temperature. Usually the plants are transplanted from seedling flats into their permanent location in the greenhouse, when they are 4 to 8 in. high. If they could be transplanted in a later stage, when they have reached a height of 3 ft, a considerable fuel saving could be effected. Such later transplanting is now feasible, if the plants have been sprayed with a 10 percent sugar solution 1 to 3 days prior to transplanting.

Ripening of Tomatoes: Fig. 2 shows the relationship between night temperature and total production in the Ailsa Craig variety. The plants were pruned back to a single stem, and all were topped above the fifth inflorescence. Therefore, production per plant could have been higher if they had not been topped. However, the plants had to be moved from one greenhouse to another to obtain the different day-and-night temperature combinations, and thus growth had to be curtailed when the plants became higher than the doors through which

they had to pass. This figure does not show the progress of ripening, which was much faster at the higher temperatures. Since earliness of production is an important factor in greenhouse tomato production, Fig. 3 shows the total weight of fruits harvested as a function of time. It has to be realized that this production was obtained in spring and summer under very favorable light conditions.

Day Temperature: Day temperature is much more important in tomato production than had been realized in earlier experiments. A marked optimum occurs at 73 F; at temperatures both higher and lower, production is substantially less. This makes an excellent argument for air conditioning of greenhouses. Whereas night temperatures can be sufficiently controlled by old-fashioned heating devices, day temperatures can be controlled effectively only by air conditioning. It is obvious that higher temperatures per se are not injurious, and therefore tem-

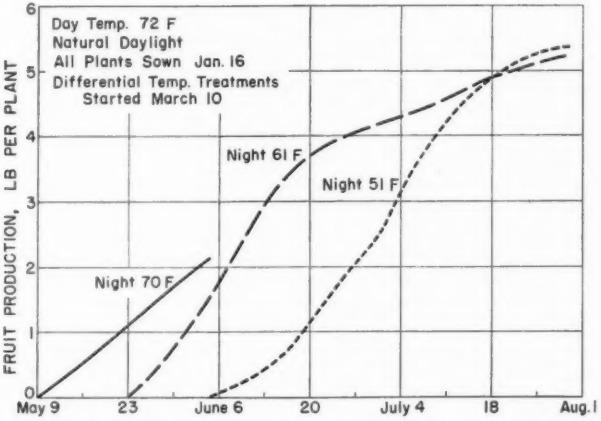


FIG. 3. RELATIONSHIP BETWEEN NIGHT TEMPERATURE, TOTAL FRUIT PRODUCTION, AND RIPENING TIME FOR THE AILSA CRAIG TOMATO

porary rises in temperature inside the greenhouse are tolerated. In most areas wet bulb temperatures are such that with an air change 60-100 times per hr evaporative cooling suffices to keep greenhouse temperatures down to 75 F most of the time; occasional rises to 85 F will occur but are not likely to cause damage. With such a rate of air change, no white-washing of the greenhouse roof is necessary, and thus full advantage can be taken of all sunlight.

INFLUENCE OF PRECIPITATION

A few preliminary tests have been carried out to find whether rain influences tomato growth. It turned out that daily sprinkling of the leaves with water (to simulate rain) would decrease their growth in comparison with plants watered only through the soil. This indicates that the proper way of watering in a tomato greenhouse is through subirrigation.

ECONOMICS OF AIR CONDITIONING

The question as to whether air conditioning can be economically justified in tomato growing can be answered with the following facts in mind. In ordinary greenhouses in England and the Netherlands a production of 40 to 50 tons of fruit per acre is normal. When extra care is taken in all growing operations, such as pest and disease control, fertilizing, and watering, production has been pushed to 80 to 90 tons per acre (at Plant Protection, Ltd., Fernhurst, England, G. Locke in charge). From the production figures under controlled temperatures it can now be calculated that 160 tons per acre should be attainable. Therefore, it seems as if air conditioning by itself can double production of tomatoes. With local prices, construction and labor costs, and the costs of utilities, it is possible to calculate whether the cost of air conditioning can be amortized out of increased income. These conditions will vary in different parts of the United States, since in some areas competition with outdoor tomatoes will become an important factor earlier in the year, which means abandoning the greenhouse crop earlier, and thus, in practice, the total production per acre in an air conditioned greenhouse will be less.

METHODS OF OBTAINING OPTIMAL CONDITIONS

The amount of solar radiation which reaches the inside of the greenhouse on a clear summer day is very great. This radiation is transformed into (1) heat of evaporation, (2) chemical energy, and (3) sensible heat. Process 2 (photosynthesis) does not amount to more than about 2 percent of the total radiation. Process 1 will vary much from crop to crop, but might tentatively be set at 48 percent of the radiation. This leaves another 50 percent which has to be removed largely by an air stream. When the heat radiation above 14,000 Angstrom units is removed by a water layer over the roof, then the discharge air will still be about 8 deg warmer than the incoming air at a rate of air change of 120 times per hr.

Therefore, to exert any control over the greenhouse temperature by air conditioning, the air in the enclosure has to be changed at a rate of at least 60 times per hr. This air has to move past all plant surfaces; in dead air spaces temperature will rise uncontrolled. However, the same rapid air stream is also required to supply the CO_2. Therefore, both for temperature control and for CO_2 supply a rapid air change is required.

When greenhouses are fully air conditioned, all incoming air can be filtered. This *can* eliminate not only all insects which might infest the plants, but also all smog and other toxic air-pollutants. Whereas insect-proofing of greenhouses has not been considered very much, it is quite feasible, and may be of sufficient economic value to justify it commercially.

The problem of how to introduce and circulate such large volumes of air in a greenhouse without causing air currents of excessive speed will require additional study. In the Earhart Laboratory it is solved by introducing the air through slots in the floor from a plenum just below the floor. This has worked very well, but is rather expensive, and excludes the possibility of solid benches. This may be a blessing in disguise because the solid benches in most greenhouses prevent proper air circulation between the plants causing unwanted temperature diffentials and uneven growth. The greater height of plants frequently

observed in the center of a solid bench is due to reduced circulation and higher temperatures.

Temperature Control: Temperature control in the large air conditioned greenhouses is achieved by passing the incoming air through a spray-chamber, which controls the wet bulb temperature. The spray-water temperature is regulated by a preheater in the air-stream or by injection of chilled water into the spray-nozzles. The airstream passes through reheating coils to heat the air to the required dry bulb temperature. In practice, most of the reheating is done by radiation from the sun. In the smaller air conditioned artificial light rooms, the temperature is entirely controlled by passage through two successive coils, one a heating and one a cooling coil. Pneumatically operated, modulating type controls are used throughout, and have proved entirely satisfactory.

In the latter rooms, humidity control is possible only through water-atomizers. Only in one room can a liquid absorption type dehumidifying unit control the humidity over a very wide range.

Recirculated and Fresh Air: In each system, the amount of fresh air *vs.* recirculated air can be regulated either manually or automatically. In addition to what has been said about tomatoes, two more examples will be given in which air conditioning of greenhouses may prove of economic significance. Further research in the Earhart Laboratory will undoubtedly contribute many more facts of importance in the construction of greenhouses. This is significant if greenhouses are to be made more efficient by air conditioning. The growth of orchids is very sensitive to temperature fluctuations. By air conditioning, the proper day and night temperatures can be maintained indefinitely and thus much faster and better growth is possible. In this way, for instance, Cypripediums and Phalaenopsis can be grown very successfully, using artificial light.

By proper regulation of the environment not only the growth of strawberry plants, but also the flavor of the fruit can be controlled. To get sweet strawberries, plants must be subjected to high light intensities, such as are obtainable with daylight. For their sugar production, the temperature during light exposure is rather unimportant. The typical strawberry aroma, on the other hand, develops only when the plants are subjected to light at a fairly low temperature (between 50 and 60 F). By air conditioning, it would be possible to cool greenhouses sufficiently during the night, so that the proper aroma would be produced when the plants are given artificial light during the night period.

CONCLUSIONS

Air conditioned greenhouses are essential for the investigation of plant behavior in relation to their environment. Only by such means can the effects of climate on crop plants be measured, and with the results obtained it may be possible to improve the growing conditions of many agricultural crops. Improvement might be made not only in crops which are occasionally grown in greenhouses (like tomatoes) but methods may be developed thereby, which would indicate those field conditions that might be modified so as to increase production or quality.

Several examples of the effects of controlled temperature and light duration on tomatoes, strawberries and other plants have been discussed. In the case of the tomato it was shown that under ideal growing conditions obtainable with

air conditioning a doubling of fruit production in the greenhouse is possible.

The most important factors which have to be considered in the construction of air conditioned greenhouses have been discussed.

DISCUSSION

A. J. HESS, Los Angeles, Calif.: It will be noted that the cooling water for the air conditioning system flows over the roof, thereby acting as a radiation shield. The roof acts as a cooling tower for the system.

Three air changes per minute were required, and because of the large volume of air supplied it was necessary to give special attention to air diffusion. The air was introduced successfully through a slotted floor.

It seems to me that as time goes on one of the principal uses for greenhouses will be for study of the problems concerning solar energy. While the storage of solar energy by means of plants may be inefficient, it should be remembered that plants store the energy over a relatively long period of time, and thus tend to equalize the storage regardless of the variations due to clouds, darkness or season.

Plants can be used as a means of transferring solar energy. They can be grown where solar energy is abundant, converted to chemical energy, and transported to regions having less solar energy.

B. P. FISHER, Houston, Tex.: I wish to inquire why the author has not used carbon dioxide produced by the burning of natural gas which is so abundant. An excess amount of water vapor would be produced by combustion of the gas, but could readily be eliminated if necessary.

AUTHOR'S CLOSURE: Referring to Mr. Fisher's suggestion, we have not used combustion of natural gas to produce carbon dioxide because some toxic materials might have been produced. It would seem ideal to burn gas within the duct supplying air as there would then be no heat transfer problem in disposing of the heat generated.

In our greenhouse air conditioning system the carbon filters make it possible to remove about 95 percent of the smog from Los Angeles air. The smog problem occurs in all large cities and affects animals and human beings, as well as vegetation. It is of interest that our plant research in a greenhouse should emphasize the need for removing smog from air breathed by humans and animals.



1504

No. 1504

ROOM AIR DISTRIBUTION RESEARCH FOR YEAR 'ROUND AIR CONDITIONING

Part II—Supply Outlets at Three Floor Locations

By H. E. Straub* and S. F. Gilman**, Urbana, Ill.

AN IMPORTANT problem involved in the year 'round air conditioning of residences is the distribution of heated air in winter and cooled air in summer from the same supply outlets in such a manner that satisfactory conditions of comfort are maintained at all times. This problem has been under investigation for some time at the University of Illinois in cooperation with the *American Gas Association*. The object of the research program is to obtain engineering data for selecting the proper types and locations of supply outlets for year 'round air conditioning of residences. The facilities being used for this investigation, together with the test conditions and procedure, have been described in a previous paper[1], which also presented the results of studies conducted with supply outlets at one high sidewall location.

The present paper reports the principal results obtained during the second phase of the air distribution research program which dealt with floor registers and floor diffusers at three different locations in the test room under winter heating, isothermal, and summer cooling conditions of operation.

Description of Apparatus

Since the apparatus is discussed in detail in a recent paper[1], only a brief description will be given here.

An isometric view of the test room is shown in Fig. 1. The room is 13¾ ft wide and 18 ft long, with an 8½ ft ceiling height. It is surrounded by an insulated structure, not shown in the figure, which forms corridors along the north and east walls of the room. Facilities are available for independently controlling the temperature in the corridors, in a basement space below the room, and in an attic space above the room over a range from — 10 F to 130 F. Approximately 70 thermocouples are located in the walls, attic, basement, and corridors. The north and east walls simulate *exposed* walls of a room, whereas the south and west walls simulate *inside* walls of a room.

The locations utilized for supply outlets are identified by Roman numerals. Location I, high in the west wall, was also used during 43 tests reported in the previous paper. Locations II, III, and V are in the floor: II is centered along

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** Research Assistant Professor of Mechanical Engineering, University of Illinois. Junior Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the 60th Annual Meeting of The American Society of Heating and Ventilating Engineers, Houston, Tex., January 1954.

the south (inside) wall; III is centered along the north (exposed) wall; and V is in the northeast corner at the junction of two exposed walls. The 5 × 28 in. baseboard return intake is located near the center of the west wall.

Each floor supply outlet was connected to the duct system in the basement space by the apparatus shown in Fig. 2. Supply air was delivered to a 14 × 14 in. outlet box and then directed upward into a liner by turning vanes. The liner had the same dimensions as the nominal size of the outlet, and was so located in the box that the outside edge of any supply outlet, regardless of its size, was 6 in. from the wall. The opening of the box not filled by the liner was capped

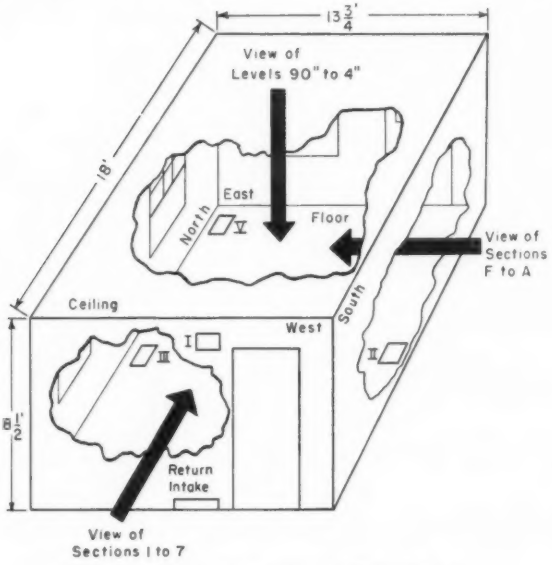


FIG. 1. ISOMETRIC VIEW OF TEST ROOM

at the floor level and sealed. Before the supply outlet was put in place, the turning vanes were adjusted to give a uniform velocity across the face of the liner. The dampers in all outlets were set for straight flow where possible; otherwise they were removed.

Fig. 3 shows a portion of the interior of the test room as viewed when looking east. The baseboard return intakes shown were not used during the studies reported in this paper. The device on wheels in the foreground is a vertical rack that can be moved to any desired position along the length of the room. The rack supports 30 heated-thermocouple anemometers for measuring air velocities and temperatures. Rows of six are uniformly spaced 4, 30, 60, 78, and 90 in. above the floor. The average temperature as measured by the central four anemometers located 30 in. above the floor, with the rack positioned in the center of the room, was used as the control (room) temperature to be maintained at 75 F during all tests.

TEST CONDITIONS AND PROCEDURE

For the majority of the winter heating and summer cooling tests, the control (room) temperature, as well as that in the basement and in the spaces outside the south and west walls, was maintained at 75 F.

During the heating tests, the temperature in the north and east corridors was held at about —5 F, and that in the attic at about 30 F. Studies were made

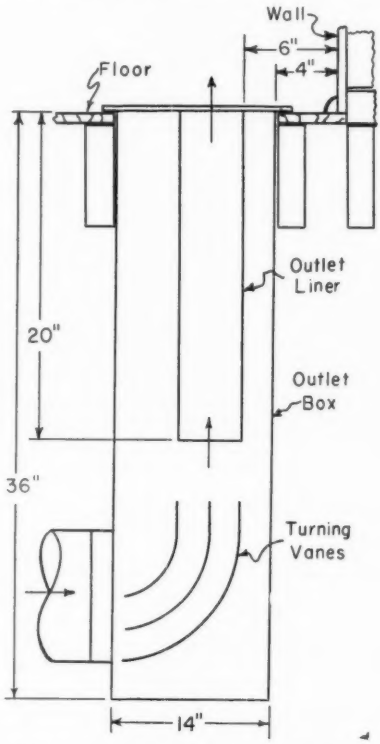


FIG. 2. SUPPLY OUTLET APPARATUS

with flow rates of 125, 230, 310, and 420 cfm (3.6, 6.5, 9, and 12 air changes per hour). The corresponding supply air temperatures were of the order of 135, 108, 99, and 93 F respectively.

During the summer cooling tests, the temperature in the north and east corridors was adjusted until the supply-air temperature was either 60 F with a flow rate of 310 cfm or 55 F with a flow rate of 230 cfm. These combinations gave 15 deg and 20 deg temperature differentials between the control and the supply air temperature. The temperature in the attic space was maintained at about 125 F.

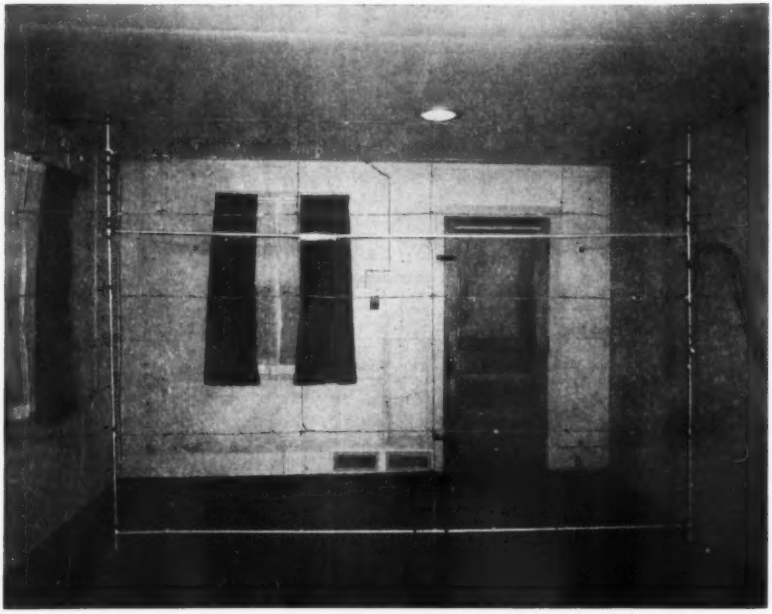


FIG. 3. VIEW OF INTERIOR OF TEST ROOM, LOOKING EAST

The sizes of the registers and diffusers were selected to provide supply air velocities, based on the outlet free area, of 300, 450, 600, and 900 fpm with the flow rates as given.

The general procedure for each test was the same as used previously[1], and consisted of velocity and temperature traverses of the room with the rack of heated-thermocouple anemometers, followed by additional explorations in the room using smoke and a portable anemometer.

TESTS CONDUCTED

The tests conducted and reported in this paper are presented in Table 1. Of the 60 tests listed, 39 were conducted under winter heating conditions, 16 under summer cooling conditions, and five were isothermal explorations. Three heating tests (Nos. 44-46) were conducted with high sidewall supply outlets, 20 with floor registers, and 16 with floor diffusers. Nine cooling tests were conducted with floor diffusers and seven with floor registers. The tests are listed in numerical order in column 1 beginning with Test No. 44. The 43 initial tests were conducted with high sidewall outlets and have been reported previously[1].

Column 2 gives the test designations. The letters preceding the numbers refer to the type of test and have the following meanings:

H = heating with a heated basement.
C = cooling.
I = isothermal.
CS = heating with a simulated crawl space temperature of 40 F.
SS = heating with storm sash.

The numbers have the following meaning: the first numeral refers to the supply outlet location designated in Fig. 1 by a Roman numeral. The second numeral refers to the nominal flow rate as follows: 0 = 125 cfm; 1 = 230 cfm; 2 = 310 cfm; 3 = 420 cfm. The third numeral refers to the nominal supply air velocity as follows: 1 = 300 fpm; 2 = 450 fpm; 3 = 600 fpm; 4 = 900 fpm. The letter D following the number signifies a floor diffuser; otherwise, a floor register was used. The test designation allows a quick comparison of test conditions. For example, Tests 67 and 68 in Table 1 are heating tests conducted

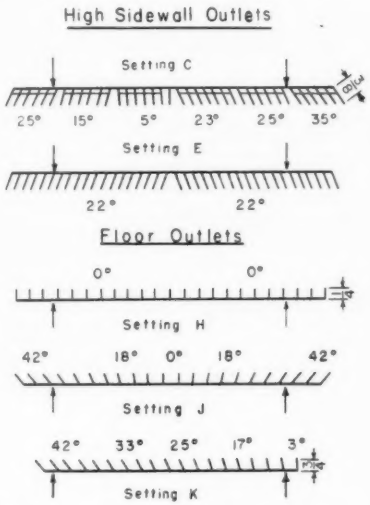


FIG. 4. DESCRIPTION OF OUTLET VANE SETTINGS

under approximately the same conditions of outlet location, flow rate, and supply-air velocity, the principal variable being the type of outlet. A register was used in No. 67 and a diffuser in No. 68.

The vane settings of Column 5 are illustrated in Fig. 4. The first two settings were used for the high sidewall studies (Nos. 44-46). The vanes for Setting H have a zero deg deflection; this setting applies to all floor registers. The remaining two settings apply to floor diffusers, with J indicating settings made at the factory and K indicating special settings made by the research personnel.

The supply-air velocity, Column 8, is the flow rate of supply air divided by the outlet free area. The air changes per hour, Column 10, are the flow rates per hour of return air divided by the cubical contents of the room. The load ratio, Column 11, is the actual cooling or heating load divided by the difference between the control temperature and the north and east corridor temperature. The velocity index, Column 12, is the percentage of the total number of traverse points in the 4, 30, 60, and 78-in. levels, excluding those within about one foot of any wall surface, at which the velocity was between 15 and 35 fpm. The temperature variation, Column 13, is the difference between the average temperature

TABLE 1—STUDIES WITH HIGH SIDEWALL AND CONVENTIONAL AND DIFFUSER FLOOR SUPPLY OUTLETS

TEST NO.	TEST DESIGNATION	OUTLET LOCATION	OUTLET SIZE	VANE SETTING	CONTROL TEMP, F	SUPPLY-AIR TEMP, F	SUPPLY-AIR VELOCITY, fpm	FLOW RATE, cfm	AIR CHANGES PER HOUR	LOAD RATIO, BTUH PER DEG F	VELOCITY INDEX, PERCENT	TEMP VARIATION 4-IN. TO 60-IN. LEVELS, DEG F
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
44	H113	I	14x8	E	75	102	505	254	7.3	107	59	13.3
45	H117	I	14x5	E	75	104	894	257	7.3	111	62	9.3
46	H121	I	14x8	C	75	109	524	228	6.5	118	52	15.5
47	H201	II	6x12	H	75	134	380	129	3.7	99	59	13.5
48	I201	II	6x12	H	73	73	348	133	2.6	0	74	0.0
49	H202	II	6x8	H	76	135	549	125	3.6	109	63	13.2
50	H203	II	4x10	H	75	136	707	127	3.6	112	65	13.1
51	H204	II	4x6	H	75	135	1210	127	3.6	110	68	11.2
52	I204	II	4x6	H	71	71	1040	123	3.5	0	92	0.0
53	H212	II	8x12	H	75	108	467	224	6.4	112	73	10.2
54	H213	II	6x12	H	75	107	628	224	6.4	108	74	9.5
55	C213 D	II	6x12	J	76	57	582	228	6.5	97	50	1.9
56	CS213 D	II	6x12	J	75	115	655	227	6.5	136	79	9.5
57	H213 D	II	6x12	J	76	106	645	227	6.5	99	83	6.6
58	H214	II	6x8	H	76	106	946	228	6.5	106	80	8.0
59	H223	II	8x12	H	75	97	634	310	8.9	103	85	6.8
60	C224	II	6x12	H	75	59	795	312	8.9	120	55	0.2
61	C224 D	II	6x12	J	75	60	813	313	8.9	123	81	0.0
62	H224	II	6x12	H	75	100	852	309	8.8	111	90	6.5
63	H234	II	8x12	H	75	97	852	415	11.8	113	76	7.2
64	I234	II	8x12	H	73	74	824	417	11.9	0	40	0.0
65	H301	III	6x12	H	75	134	374	127	3.6	106	64	13.7
66	H301 D	III	6x14	J	76	132	315	123	3.5	101	63	9.6
67	H303	III	4x10	H	75	132	700	126	3.6	106	64	12.8
68	H303 D	III	4x10	J	75	128	748	130	3.7	101	72	8.3
69	SS303 D	III	4x10	J	75	127	717	122	3.5	91	63	7.5
70	H304 D	III	2x14	J	75	134	861	125	3.6	103	72	7.9
71	C312 D	III	6x14	J	76	55	494	226	6.4	96	53	3.8
72	C313	III	6x12	H	76	56	576	228	6.5	109	89	0.1
73	H313	III	6x12	H	75	104	627	225	6.4	105	78	8.9

TABLE 1—STUDIES WITH HIGH SIDEWALL AND CONVENTIONAL AND DIFFUSER FLOOR SUPPLY OUTLETS *(concluded)*

TABLE 1—STUDIES WITH HIGH SIDEWALL AND CONVENTIONAL AND DIFFUSER FLOOR SUPPLY OUTLETS *(concluded)*

TEST NO.	TEST DESIGNATION	OUTLET LOCATION	OUTLET SIZE	VANE SETTING	CONTROL TEMP, F	SUPPLY-AIR TEMP, F	SUPPLY-AIR VELOCITY, fpm	FLOW RATE, cfm	AIR CHANGES PER HOUR	LOAD RATIO, BTUH PER DEG F	VELOCITY INDEX, PERCENT	TEMP VARIATION 4-IN. TO 60-IN. LEVELS, DEG F
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
74	SS313	III	6x12	H	75	103	632	227	6.5	98	72	8.4
75	C313 D	III	6x12	J	75	55	578	227	6.5	102	67	1.4
76	CS313 D	III	6x12	J	75	115	655	227	6.5	134	85	9.0
77	H313 D	III	6x12	J	76	107	634	223	6.4	100	85	5.7
78	I313 D	III	6x12	J	74	74	610	229	6.5	0	89	0.0
79	SS313 D	III	6x12	J	76	104	639	227	6.5	91	87	5.6
80	C314	III	6x8	H	77	56	854	228	6.5	113	83	0.1
81	C314 D	III	4x12	J	75	56	882	228	6.5	116	79	0.0
82	C316 D	III	6x12[a]	J	76	56	710	228	6.5	120	79	0.2
83	C322	III	10x14	H	75	60	400	314	9.0	127	79	0.0
84	H323	III	8x12	H	75	98	648	312	8.9	115	80	8.8
85	C323 D	III	6x14	J	75	61	668	312	8.9	126	88	0.1
86	H323 D	III	6x14	J	75	99	746	311	8.9	108	84	5.3
87	SS323 D	III	6x14	J	76	98	737	309	8.8	96	89	4.6
88	C324	III	6x12	H	75	60	802	315	9.0	133	65	0.2
89	H324	III	6x12	H	75	98	850	310	8.8	103	84	6.3
90	C324 D	III	6x12	J	75	60	796	312	8.9	121	90	0.2
91	H324 D	III	6x12	J	75	99	869	311	8.9	104	79	4.5
92	I324 D	III	6x12	J	73	73	837	314	8.9	0	74	0.0
93	H501	V	6x12	H	75	138	375	127	3.6	115	64	13.9
94	H511	V	12x12	H	75	105	319	226	6.4	108	63	11.1
95	H513	V	6x12	H	75	105	638	229	6.5	105	79	9.7
96	H513 D	V	6x12	K	75	105	742	227	6.5	100	87	7.4
97	H515 D	V	(4x10 (2¼x14	K) K)	76	108	730	227	6.5	95	63	7.1
98	C522	V	12x12	H	75	62	402	311	8.9	114	56	1.7
99	H522	V	12x12	H	75	96	428	310	8.8	107	78	9.7
100	C524	V	6x12	H	75	60	801	314	9.0	126	34	0.2
101	C524 D	V	6x12	K	75	59	945	315	9.0	137	51	0.3
102	H524	V	6x12	H	75	99	855	312	8.9	103	71	7.9
103	H533	V	12x12	H	76	92	561	411	11.7	99	86	8.4

[a] Part of free area sealed off to provide 710 fpm velocity.

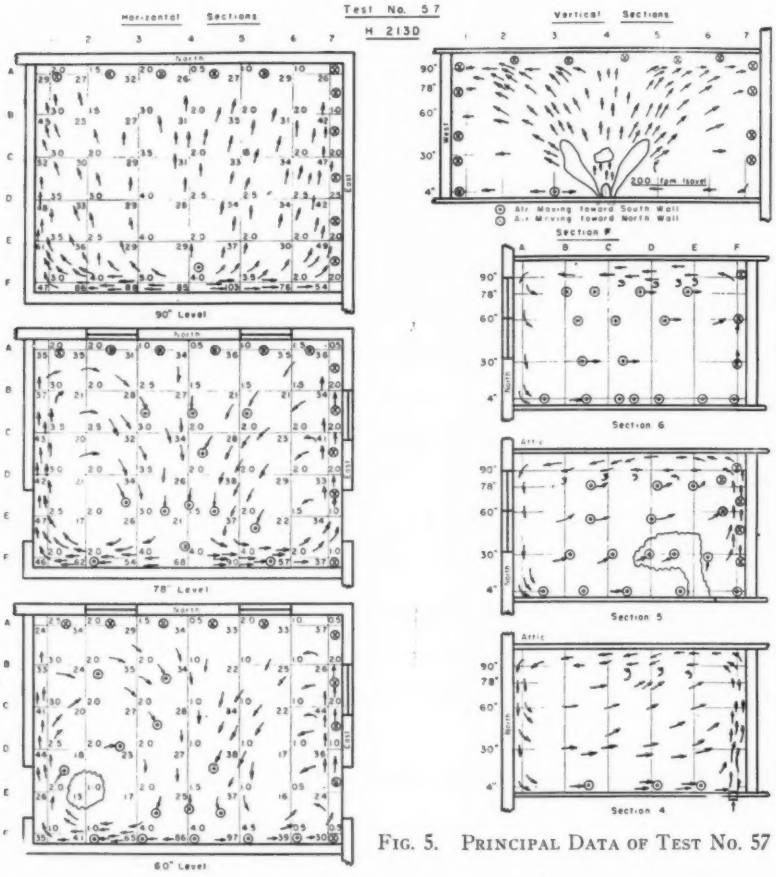


FIG. 5. PRINCIPAL DATA OF TEST NO. 57

at the 60-in. level and that at the 4-in. level, and for the heating tests the values are adjusted to a 0 F outdoor temperature.

WINTER HEATING STUDIES WITH A HEATED BASEMENT

Of the tests listed in Table 1, 33 were conducted under the conditions of winter heating with a heated basement; *i.e.,* basement space temperature of 75 F. During these studies, the *outdoor* temperature (north and east corridor temperature) was maintained at approximately —5 F.

For each test, the principal data for analysis consisted of the temperatures and velocities at 210 points in the room obtained by traverses with the 30 heated-thermocouple anemometers, together with the results of additional explorations of the jet characteristics and the general motion of the air masses within the

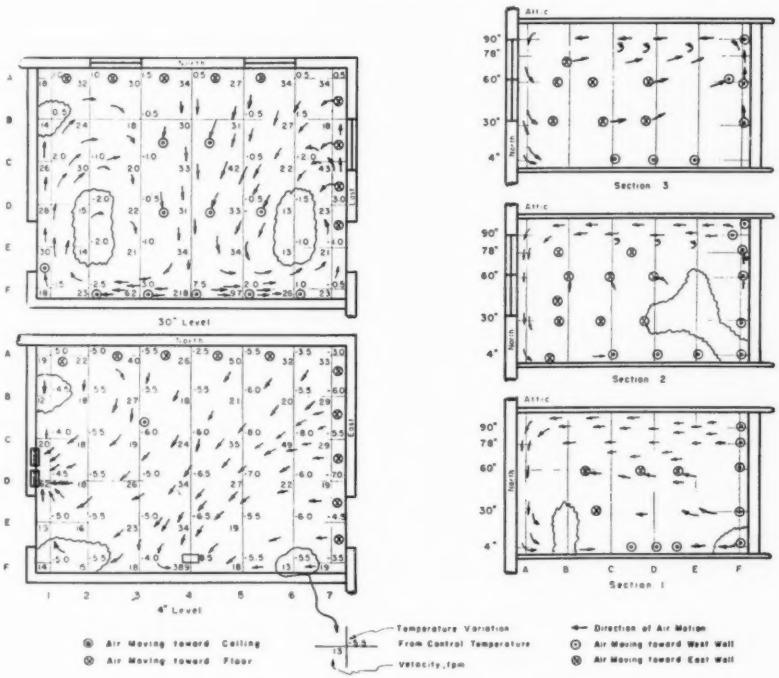


FIG. 5 *(Cont'd)*. PRINCIPAL DATA OF TEST NO. 57

room. Fig. 5 is representative of the general method of compiling the principal data obtained for each test. Such diagrams, combined with supplementary data, served as the basis for analyzing and interpreting the data from all tests.

The diagrams in the left column of Fig. 5 show five horizontal sections at the 90, 78, 60, 30, and 4-in. levels of the room as viewed from above (see vertical arrow in Fig. 1). Each intersection of the grid lines in the diagrams represents a point at which the velocity and temperature were measured; the upper number indicates the temperature variation from the control (room) temperature, and the lower number indicates the velocity. For example, at the 4-in. level at F-6, the temperature is 5.5 deg below the control temperature, and the velocity is 13 fpm. Where no temperature value is given, the variation is zero. The arrows show the direction of the air motion at the particular level.

The diagrams in the right column of Fig. 5 show seven vertical sections of the room. Sections designated by the numerals 1, 2, etc. lie along the correspondingly numbered lines on the diagrams in the left column (see oblique arrow in Fig. 1). Sections designated by the letters A through F (only section F is shown) lie along the correspondingly lettered lines of the diagrams in the left column (see horizontal arrow in Fig. 1).

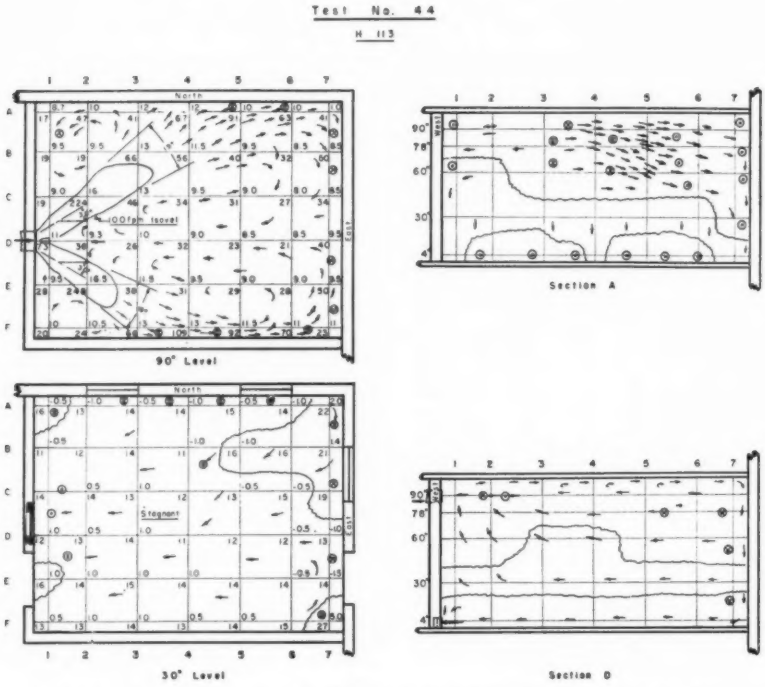


FIG. 6. PRINCIPAL DATA OF TEST NO. 44 *(Compare with Test No. 73)*

Although a total of 13 vertical and 5 horizontal sections were used in analyzing the data, diagrams of some of the less significant sections have been omitted from this paper. When all 13 vertical sections are fixed in upright positions and the 5 horizontal sections sandwiched into these in an egg-crate manner, a three-dimensional representation of the temperature variations, velocities, and flow patterns in the room is obtained.

Fig. 5 shows the principal data for Test 57 which was conducted with a 6 in. × 12 in. diffuser in Location II, a nominal flow rate of 230 cfm, and a nominal supply-air velocity of 600 fpm. From Columns 12 and 13 of Table 1 the velocity index is 83 and the temperature variation 6.6 deg.

Section F in the upper right of the figure shows that the primary air splits into two main jets, with axis about 40 deg from the register centerline, and a minor jet straight up. The mixture of primary and induced air; *i.e.*, the total air, fans out and blankets the upper portion of the south wall, and then flows along the ceiling (Note 90-in. level diagram) and along the east and west walls. At the 78, 60, and 30-in. levels, two circulatory motions are set up as a partial result of the induction effect of the supply jet. At the 4-in. level, the cooler

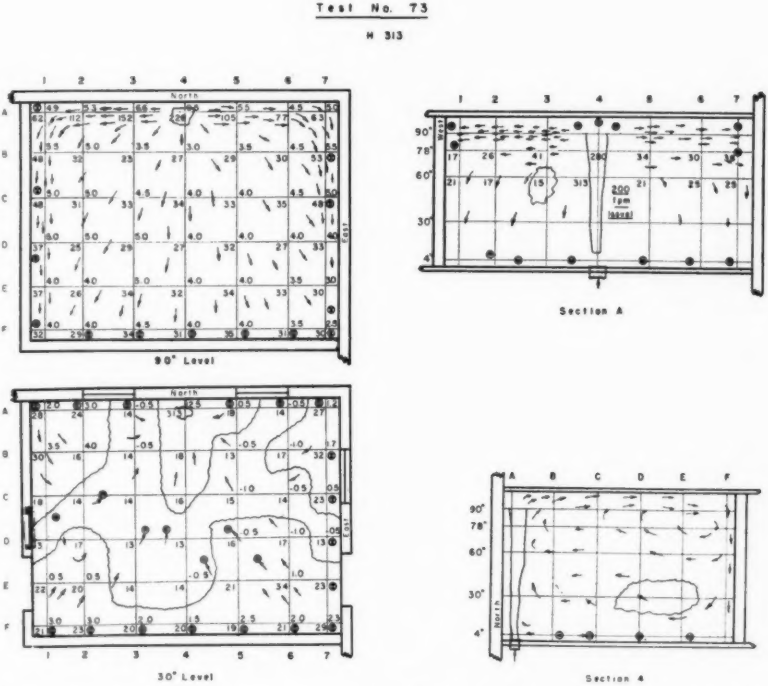


FIG. 6 *(Cont'd)*. PRINCIPAL DATA OF TEST NO. 73 *(Compare with Test No. 44)*

air falling off the inside surfaces of the exposed walls flows in the general direction of the warm corner near F-1, although some is drawn into the return and some is induced by the supply jet. In vertical planes, as shown in Sections 1 to 6, the general motion is up the south wall, across the ceiling, and down the north wall.

Regions in which the velocity is 15 fpm or less are enclosed by a jagged line. Within these regions a stagnation condition exists in that the air motion is hardly perceptible and smoke diffuses into a layer and remains there for some time. Since the HEATING VENTILATING AIR CONDITIONING GUIDE, 1953,[2] states that air velocities less than 15 fpm give people a feeling of stagnation, the air distribution within these regions is considered to be poor. Hence, the number and size of such regions provide one index of the quality of the air distribution within a space. A second index is the temperature variations within the zone of occupancy, while a third is the size of the region in which the velocities are in the desirable range of 15 to 35 fpm.

Fig. 6 shows four diagrams for each of two heating tests conducted under conditions nearly equivalent to those of Test No. 57 (Fig. 5) with the exception of the location and type of supply outlet.

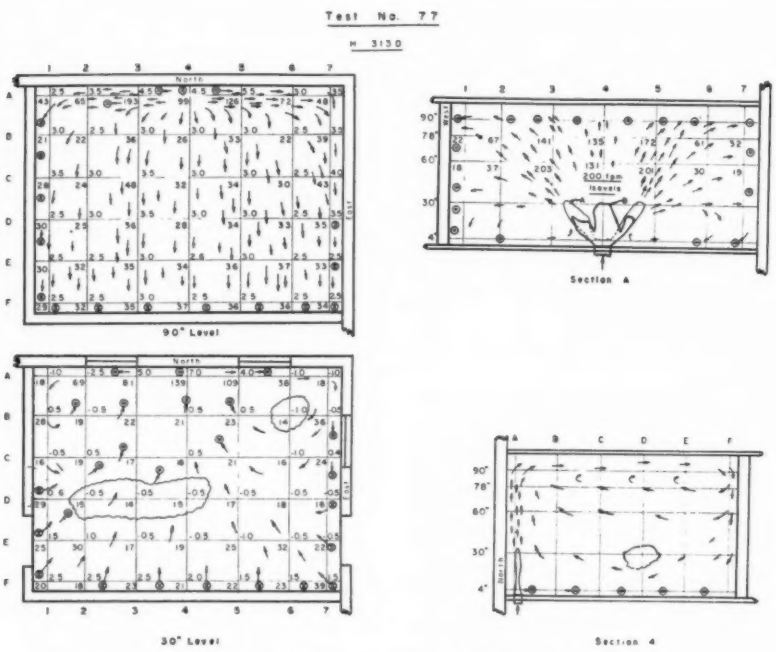


FIG. 7. PRINCIPAL DATA OF TEST NO. 77

Test 44 was conducted with a high sidewall supply outlet at Location I, and the diagram of the 90-in. level in Fig. 6 shows the two jets produced. This two-jet condition was discussed in the previous paper.[1] The flow pattern here at a level above the zone of occupancy is two circulatory motions, one clockwise and one counterclockwise. Reference to the large stagnation regions shown in the other diagrams indicates that practically all of the induction and air motion took place above the occupied zone. Such a condition was a result of the buoyancy effect of the warm supply air issuing from an outlet located above the occupied zone. The large stagnation region, together with the 13.3 deg temperature variation (Col. 13, Table 1), indicates poor air distribution.

The lower half of Fig. 6 shows four diagrams from an equivalent heating test with a conventional floor register at Location III. Referring to the Section A diagram of Test No. 73, the upward discharge of the air from the register, together with the natural buoyancy of the warm supply air, results in little spread of the jet and a relatively small amount of induction before the jet impinges on the ceiling. The stream then flows out across the ceiling (90-in. level diagram) and down the inside wall (Section 4).

Comparison of the diagrams for Tests Nos. 44 and 73 in Fig. 6 shows that the air temperature near the ceiling is lower with the floor register (Test No. 73) than with the high sidewall register (compare 90-in. level data). This re-

sults in a lower temperature variation in the occupied zone, as reference to Table 1 will show. In addition, the stagnation region is much less for Test No. 73 than for Test No. 44, although nearly all velocities in the former are less than 35 fpm and therefore not excessive. Consequently, it is concluded that the air distribution from the conventional floor register at Location III, Test No. 73, is better than that with the high sidewall register at Location I, Test No. 44, for the particular test conditions used.

Similar analyses made of comparative tests at various other outlet locations corroborated the conclusion that conventional floor registers can be expected to provide better air distribution during winter heating than high sidewall registers.

Fig. 7 presents four diagrams from a heating test under the same conditions as Test No. 73, Fig. 6, except that the outlet is a floor diffuser instead of a floor register. Comparison of the diagrams and the Table I data shows that with the diffuser (Fig. 7) the size of the stagnation region is considerably reduced and the temperature variation is 30 percent less. Moreover, analysis of all 13 diagrams showed a greater uniformity of velocity throughout the occupied zone. From the results of this and other tests, it was concluded that diffusers furnished better air distribution than registers during winter heating. The principal reason for this is that the wide spread of the jet issuing from a diffuser provides a large jet surface which induces considerable room air after only a short travel. This then causes rapid mixing of the warm supply air with room air and subsequent small temperature variations in the occupied zone.

It should be noted that good air distribution can be obtained with a diffuser on an inside wall as well as on an exposed wall. This can be verified by comparing Figs. 5 and 7, in which the only variable is the location of the diffuser. Little difference in the air distribution can be noted in the diagrams. However, detailed comparison of the test data indicated that the diffuser located along the exposed walls provided slightly better air distribution. The principal advantage of the exposed wall location for diffusers is that the issuing jet effectively counteracts the cold air falling down the walls and windows (see Sections A and 4 of Fig. 7). However, even the diffusers located along the inside wall provided much better air distribution than any of the conventional registers in any of the three floor locations shown in Fig. 1.

For a given diffuser, the results of the studies showed that the order of best-to-worse location was III, II, and V. Referring to Fig. 1, note that V is a corner location. Such locations have distinct disadvantages because the adjacent wall surfaces so confine the jet that little induction of room air is possible, the result being large temperature variations and little air motion in the occupied zone above 15 fpm. Test No. 97 was conducted in an attempt to improve the air distribution from a corner location. Two diffusers were butted together, with one facing along the east wall and the other facing along the north wall. Special vane settings were used in an effort to blanket both the east and north walls with warm air. However, little improvement in the air distribution was obtained. From this and other tests conducted at Location V, it was concluded that restricted regions such as corners are not desirable for diffusers.

For a given floor register, the results of the studies indicated that Locations II, III, and V resulted in about the same air distribution which, however, was better than that obtained with high sidewall registers at Location I. On the other hand, floor registers resulted in poorer air distribution than obtained with

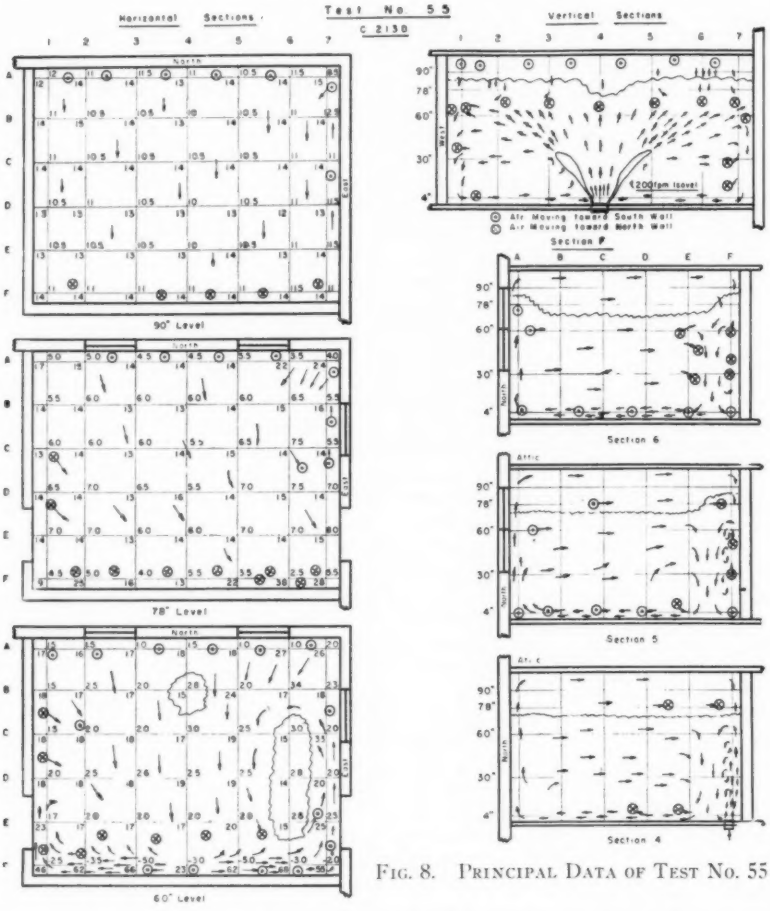


FIG. 8. PRINCIPAL DATA OF TEST NO. 55

diffusers at Location V. In general then, the floor diffusers provided better air distribution than the floor registers, which in turn provided better air distribution than the high sidewall outlets.

During the course of conducting tests with registers or diffusers at each of the four locations, the supply-air velocity was varied by installing different sizes of outlets. Also, with a particular outlet installed, tests were conducted at from one to four rates of flow (cfm). The results of these studies showed the individual effects of supply-air velocity and flow rate on the air distribution. In general, the flow patterns illustrated in the foregoing are representative of those at other velocities and flow rates, the principal exception being the combination of the lowest velocity and smallest flow rate. For this exception, since the supply

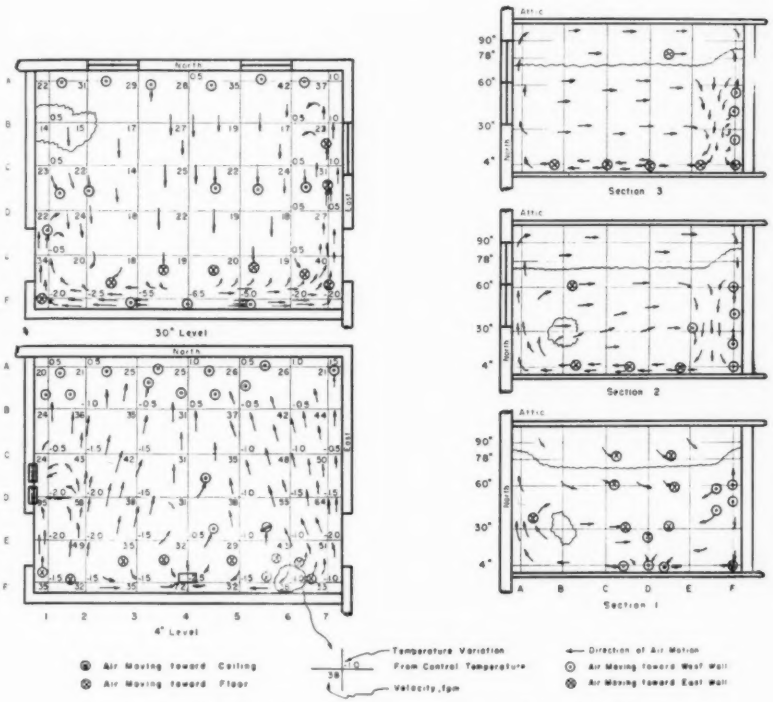


FIG. 8 *(Cont'd)*. PRINCIPAL DATA OF TEST NO. 55

air had relatively little energy, it floated to the ceiling and spread out across it in a gentle motion. As a consequence, the room air motion was primarily a result of free convection currents along the surfaces. The usual downward motion on the opposite wall, *e.g.*, Section 4 of Fig. 7, was therefore not nearly as pronounced as in the other tests.

These velocity and flow rate studies showed that for the better floor locations (II and III) a change in supply-air velocity had little effect on the air distribution. However, a change in flow rate, with its consequent change in the temperature of the supply air, had a relatively large effect on the air distribution, which was better at the higher flow rates. Results of the three tests with high sidewall outlets, however, indicated that either an increase in supply-air velocity or an increase in flow rate produced a significant improvement in the air distribution.

HEATING WITH SPECIAL TEST CONDITIONS

Of the tests listed in Table 1, six were conducted under special conditions to evaluate the effect of: (1) reduced basement space temperature; and (2) storm sash on the windows.

Tests Nos. 56 and 76 were conducted with a 40 F basement space temperature to simulate an unheated crawl-space beneath a room. The results were then compared with those of equivalent tests with a heated basement, Tests Nos. 57 and 77. For the crawl-space condition, the temperature variations between the 4-in. and 60-in. levels were about 3 deg higher, the floor surface temperatures were about 10 deg lower and, of course, the heating load was greater. In other respects the air distribution was about equivalent for the two cases. Therefore, comparable air distribution in crawl-space applications evidently can be obtained by keeping the floor surface temperature as high as possible, a condition obtained, for instance, by properly insulating the floor.

The effect of storm sash was evaluated by repeating Tests Nos. 68, 73, 77, and 86 with storm sash installed. The results of these tests (Nos. 69, 74, 79, and 87) indicated that the air distribution was only slightly improved by the addition of storm sash. Referring to Col. 13 of Table 1, the sash resulted in only slight reductions in temperature variations in the room. Other than this, the effect was limited to some small reductions in the downward convection currents directly beneath the windows. From these studies it was concluded that while storm sash is desirable for reducing the heating load, it is not necessarily essential to obtain good air distribution in a room under winter heating conditions of operation.

SUMMER COOLING STUDIES

After analyzing the results of the winter heating studies, 16 tests were conducted under summer cooling conditions. Of these, 10 were conducted at the location which had provided the best air distribution during heating, Location III, and three each were conducted at Locations II and V.

Representative results for the cooling studies are shown by the diagrams for Test No. 55 in Fig. 8. The nominal supply-air velocity was 600 fpm and the nominal flow rate was 230 cfm, with a corresponding temperature differential between the supply and room air of 20 deg. This test and Test No. 57 in Fig. 5 provide an example of a year 'round system utilizing the same flow rate for winter heating and summer cooling.

Section F shows that the characteristics of the jets are very similar to those in the heating test, but the total air does not go above the 78-in. level. Sections 1 to 6 show that the buoyancy effect of the cool total air causes it to leave the jet and fall to the floor, resulting in a general motion across the floor, up the warm north wall, and back to the south wall at all levels above 4 in. The small temperature variations at the 4-in. level indicate that mixing of the jet and room air was very complete. A stagnation region is seen to exist from about the 70-in. level to the ceiling. This region has very high temperature variations, as shown by the diagrams of the 78-in. and 90-in. levels. Test No. 75 was conducted with Location III under the same conditions as Test No. 55 with Location II and the same general air motion was found for both tests.

Because of the stagnation region found in Tests Nos. 55 and 75, the level at which the stagnation region began was determined for various supply-air velocities. Hence, Tests Nos. 71, 81, and 82 had the same location and flow rate as Test No. 75, but had supply-air velocities of 494, 882, and 710 fpm. During Test No. 71 the total air was carried to the 60-in. level, and a stagnation region existed from this level to the ceiling; whereas, in Tests Nos. 81 and 82 the total air was carried to the ceiling and no stagnation region existed. From these

tests it was determined that the minimum supply air velocity required to prevent a stagnation region occurring was of the order of 700 fpm for the floor diffusers.

Test No. 72 utilized a register with about the same flow rate and supply-air velocity as Test No. 75, which utilized a diffuser. No stagnation region was formed and the air distribution was good. Increasing the supply-air velocity, Test No. 80, also resulted in good air distribution.

By comparison of the data for the tests with both diffusers and registers in Location III, it was concluded that good air distribution can be obtained with a 20 deg differential with both floor registers and floor diffusers; also, for the optimum air distribution the required supply-air velocity is somewhat less for a register than for a diffuser. As previously reported[1], good air distribution was not obtained with the high sidewall outlets (Location I) and a 20 deg differential.

Tests Nos. 83, 85, 88, and 90, which were conducted with a flow rate of about 310 cfm and a temperature differential of 15 deg, included diffusers and floor registers as well as variable supply-air velocity. The temperature variations were very small for all of these tests. However, with a register and the higher supply-air velocity, the room air motion was excessively high. Also, the diffusers gave a more uniform room air motion than registers.

Comparison of the results obtained with supply outlets at three different floor locations indicate that the order of preference is Location III, II, and V. Therefore, a floor outlet located at the center of an exposed wall (Location III) evidently has excellent posibilities of providing good air distribution during year 'round air conditioning.

ISOTHERMAL STUDIES

Five tests were conducted under isothermal conditions with floor outlets at Locations II and III, and a portion of the principal data of one test is shown in Fig. 9. This test was conducted under the same flow conditions as the winter heating test of Fig. 7. Comparison of these data shows that the jet characteristics are the same for both. In addition, the room air motion, although greater in magnitude during the isothermal test, has a pattern nearly similar to that of the heating test. Such similar characteristics of isothermal and heating tests were also obtained during the other four isothermal tests. Hence, with floor outlets, under many conditions the room air motion during isothermal conditions will be representative of that which would be obtained under winter heating conditions. In contrast, the previous studies with high sidewall outlets indicated that the room air motion during isothermal conditions was representative of that obtained during the summer cooling conditions of operation.

With respect to a comparison of isothermal and summer cooling data with floor outlets in use, the room air motion was found to be significantly different. Even during the studies utilizing a high flow rate (small temperature differential) and a high supply-air velocity, gravitational forces had considerable influence on the flow pattern. Consequently, instead of a general motion across the ceiling and down the far wall that occurred under isothermal conditions, under summer cooling conditions of operation the air stream dropped into the occupied zone.

From these and previous isothermal studies it is concluded that, although under certain conditions isothermal and non-isothermal test conditions will yield

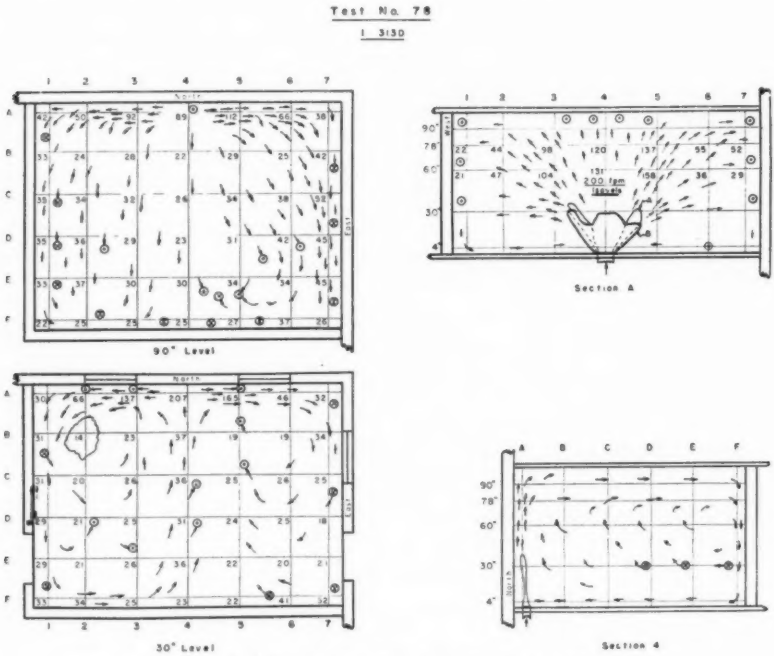


FIG. 9. PRINCIPAL DATA OF TEST NO. 78

similar air motion patterns in a room, the results of isothermal studies cannot generally be interpreted as predictions of the air motion in a room under either winter heating or summer cooling conditions of operation.

SUMMARY OF RESULTS AND CONCLUSIONS

1. During year 'round conditions of operation, better air distribution was obtained with floor outlets than with high sidewall outlets.

2. Floor diffusers produced a more uniform air motion and a lower temperature variation during heating than floor registers. Diffusers were especially effective in the unrestricted location near the center of the exposed wall.

3. With floor outlets and winter heating conditions of operation, an increase in supply-air velocity produced only a slight improvement in the air distribution. However, an increase in flow rate, with a subsequent lower supply-air temperature, resulted in a significant improvement. With a high sidewall outlet, either an increase in the supply-air velocity or the flow rate produced a significant improvement.

4. The best air distribution during cooling was obtained with outlets located near the center of the exposed wall. Under these conditions good air distribution was obtained with a temperature differential as large as 20 deg.

5. To obtain optimum air distribution during cooling, a higher supply-air velocity was required with floor diffusers than with floor registers.

6. For the simulated crawl-space conditions, the floor surface temperature was much lower and the temperature variation somewhat higher than those obtained under comparable heated-basement conditions.

7. Storm sash installed on the windows obtained only slight improvement in the air distribution.

8. Results obtained during isothermal studies cannot generally be interpreted as predictions of the air motion in a room under either winter heating or summer cooling conditions of operation.

9. A floor outlet located at the center of an exposed wall has excellent possibilities of providing good air distribution during year 'round air conditioning, particularly if the flow rate used during summer cooling is also used during winter heating.

ACKNOWLEDGMENT

This paper is a report of one phase of a research program being conducted under the terms of a cooperative research agreement between the *American Gas Association* (PAR Plan, *A.G.A.* Project DGR-2-AC) and the Engineering Experiment Station of the University of Illinois. The broad objective of this research program, which was initiated in October 1947, is the improvement of air distribution systems for year 'round air conditioning. The material presented in this paper will ultimately comprise part of a University of Illinois Engineering Experiment Station Bulletin.

Acknowledgment is made for the advice and counsel rendered by the *A.G.A.* Technical Advisory Group for Heating and Air Conditioning Research, and especially for the help and suggestions of W. F. Friend, H. C. Pierce, and G. F. Zellhoefer, Technical Advisors for the project.

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DISCUSSION

E. R. AMBROSE, New York, N. Y. (WRITTEN): The authors and their associates are to be congratulated for devoting the necessary time and effort to present the data and results in such a useful and informative manner. It is gratifying to know that the project will be continued with an examination of baseboard slot-type outlet and low sidewall diffusers.

The effect of the return location on the air distribution was not mentioned in the paper. In the previous article on the subject, however, a gentle flow towards the return, below the 4-in. level, was observed but it was stated that the two tests were not sufficient to warrant a definite conclusion. Were any additional observations made or is any further work planned on the location of the return?

The effect of the grille aspect ratio on the air distribution was not specifically mentioned. Were there any indications that a high aspect ratio (of say 4 to 5 or more) would improve the circulation of a high sidewall outlet? The natural tendency would be to locate two smaller high sidewall grilles on the inside south wall instead

of one larger outlet on the west wall in order to shorten the throw as well as to improve the possibility of a better coverage. Is it planned to investigate other sidewall locations as well as to explore the use of ceiling outlets and the special sidewall outlets now on the market?

All of the data in the paper were taken in a room bare of furniture and with continuous fan operation. Would the results be changed materially with conventional furnishings using intermittent fan operation?

It was concluded in the paper that the floor outlet, located at the center of an exposed wall, would provide the best distribution for year 'round air conditioning but a location adjacent to an inside wall (south wall in the tests) might also be satisfactory. Is it the intent in future tests to explore the possibility of locating both the supply and return openings in, or adjacent to, an inside wall for the purpose of materially lowering the first cost of the system?

It is realized that, in a project of this nature, there is a practical limit to the various possibilities which can be investigated. It is hoped that the authors can give some answers to the questions based on their observations to date.

W. O. HUEBNER, New York, N. Y. (WRITTEN): The warm air heating system plays an important role among the various heating systems used in the private homes of this country. One advantage is that it may readily be adapted to serve for cooling as well as for heating.

With cooling added to heating the necessity for good air distribution in homes is obvious. The investigations at the University of Illinois, are, therefore, of great interest to all concerned with the air conditioning of homes.

An entirely practical approach is used in the investigations—similar to the approach used by the manufacturers of air distribution devices—and consists in simulating actual conditions in a room as closely as possible. The basic studies of free ventilating jets made at the A.S.H.V.E. Research Laboratory and at Case Institute of Technology in Cleveland are thus happily complemented.

The University of Illinois' program has already yielded valuable contributions to the solution of practical air distribution problems in residences, and this paper adds to our knowledge. Its careful study is most rewarding, and the methods used by the authors in tabulating their measurements might well serve as a guide to others. I refer particularly to the method used in indicating by letters the type of test made, thus easily identifying a particular investigation and making it easy to compare different tests. The analysis of the tests, as well as the whole representation, is to be commended.

Some of the results obtained, particularly those regarding the good performance of floor outlets, are of special interest. The shortcomings of floor outlets are, of course, well-known and the question whether or not it is advisable to install them in homes may be debated at length.

Generally, the results of the authors confirm the findings of the air distribution specialist. The superior performance of diffusers is noted and confirms the experience that an air distribution device, in order to perform satisfactorily must readily entrain a large amount of room air, and create an unceasing air movement in the room without permitting stagnation of the air.

As an engineer having particular interest in air distribution progress, I should like to add the following remarks:

Investigators in air distribution research should make more use of each other's findings. It would be of great interest, for example, to see the formulas developed by Prof. G. L. Tuve and his associates of Case Institute of Technology applied to the research of the University of Illinois. These formulas, thanks to G. E. McElroy, Pittsburgh, Pa., appear in chart form in THE GUIDE, 1954.

The members of the Society should take enough interest in these problems to help the Society continue air distribution research at its Research Laboratory. I am happy

to add that Prof. Alfred Koestel at Case Institute of Technology is continuing his investigations.

C. M. ASHLEY, Syracuse, N. Y.: I think the most significant conclusion from this very interesting study is the fact that even the best of the arrangements still leave a very substantial temperature stratification between the floor level and the breathing level. This indicates that there is a need for one of two measures: Either the windows and the bulk of the wall must be better blanketed with a stream of air to prevent the cold air from dropping to the floor, or conversely, air must be picked up off the floor and mixed with the supply stream. It would be most interesting indeed to learn how successfully either one of those two methods can be made in a future research study.

I think it would be helpful to also study the performance of outlets located directly under the windows which in many cases today is a situation which obtains in practice.

Of particular interest are the comments that storm sash obtained only slight improvement in the air distribution. When one takes into account the effect of the mean end radiant temperature, the difference between single and double sash becomes quite significant.

Concerning the effect of the lowered crawl space temperature, I would like to note that one of the opportunities which exists in connection with a crawl space type of house is not only to prevent the lowering of temperature but by introducing some heat into the crawl space to use it in effect as a radiant heating panel; my own observations have been that that is very effective. Looking at the results shown in this paper and projecting them to a still higher temperature would indicate that that might make even an overhead type of air distribution much more palatable than it is otherwise.

AUTHORS' CLOSURE (H. E. Straub): In regard to Mr. Ambrose's remarks, he has very aptly outlined our future program. We plan future work on baseboard type outlets with vertical and horizontal slots, side wall diffusers, high side wall outlets on outside and inside locations, ceiling outlets and also multiple high sidewall outlets on the inside wall as suggested by Mr. Ambrose.

A portion of the studies with baseboard and low sidewall diffuser outlets will be conducted with the outlets located under the windows as Mr. Ashley suggested.

Since this paper was prepared, additional data pertaining to the effects of return location have been obtained and still more are planned. The future studies will include tests with returns in baseboard, floor, and high sidewall locations.

During our studies the aspect ratio of the high sidewall varied from about 1.8 to 2.9, and of the floor outlets from about 1 to 7. These variations are all that might be expected with residential type outlets. Therefore, as in the curves shown today, no attempt has been made with the data to separate the effect of aspect ratio from the other variables.

In regard to the question of furniture, the diagrams that were presented in the paper and also those today show the room air motion without furniture. If a chair were placed in the center of the room, the air would attempt to follow the path obtained when the chair was not there. However, the air would be deflected, and I think that the resulting room air motion could be easily deduced. This contention is supported by the fact that when the investigators were in the room, a sudden movement by the investigators would disturb the room air motion, but within a few seconds it would return to its original flow pattern.

The indications shown by our data should not be changed by intermittent operation of the fan. Our studies were conducted with steady-state conditions which included continuous fan operation. During steady-state conditions the supply air temperature remains high and near a maximum. During intermittent operation the burner is cut-off before the fan, and thus the supply air temperature is relatively low near the

end of the on-cycle of the fan. Therefore, the system approaches isothermal conditions in which the supply-air and room-air temperatures are the same. It is quite probable then that the steady-state conditions of our tests would yield higher temperature variations than would be experienced in a residence. However, this is desirable from a test standpoint as the differences in performance are magnified and more easily recognized.

We appreciate the comments made by Mr. Huebner. I might add, that I also would like to see this work correlated with the analytical approach being made at the Case Institute of Technology.

We also thank Mr. Ashley for his comments and we will certainly take into account the suggestions regarding these investigations.

No. 1505

COOLING A SMALL RESIDENCE USING A PERIMETER-LOOP DUCT SYSTEM†

By D. R. BAHNFLETH,* C. F. CHEN,** AND H. T. GILKEY***

PREVIOUS INVESTIGATIONS[1-9] in summer cooling were conducted in Warm Air Heating Research Residence No. 1 during the summers of 1932 through 1938, and 1940. During the summer of 1952 a series of summer cooling investigations[10] was begun in Warm Air Heating Research Residence No. 2. Both of these residences were of tight frame construction and had basements. Because of the increasing use of year 'round air conditioning in homes equipped with perimeter heating systems, it seemed advisable to begin summer cooling studies in Research Residence No. 3, which is a low cost, basementless residence with a concrete slab floor.

The principal objectives of this investigation were:

1. Determination of the cooling load and its hourly variation when cooling Research Residence No. 3 under both day and night conditions.
2. Determination of the time lags in heat flow through the walls and ceiling.
3. Comparison of the actual cooling load with the cooling load calculated according to the procedure outlined in THE GUIDE 1952.[11]
4. Determination of the characteristics of the warm-air perimeter-loop system when used as a distribution system for summer air conditioning.
5. Determination of the operating characteristics of the mechanical condensing unit installed in the Residence.

DESCRIPTION OF RESIDENCE

Residence No. 3 is a single-story, low cost home with a concrete slab floor. It is of standard frame construction, and has a relatively large amount of glass area. The south exposure is shown in Fig. 1. The walls are uninsulated, but the ceiling is insulated with 3⅝ in. thick bat-type mineral wool insulation. The exposed wall section consists of a double course of cedar shingles, building paper, shiplap sheathing on 2-in. × 4-in. studs, ½-in. gypsum board with aluminum foil backing and a simulated plaster finish. The attic is vented by louvered openings in the gable ends. The calculated coefficients of heat transmission U for the walls and ceiling are 0.21 and 0.07 Btu per (hr) (sq ft), respectively.

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† This investigation was part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association*.

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[1] Exponent numerals refer to References.

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end of the on-cycle of the fan. Therefore, the system approaches isothermal conditions in which the supply-air and room-air temperatures are the same. It is quite probable then that the steady-state conditions of our tests would yield higher temperature variations than would be experienced in a residence. However, this is desirable from a test standpoint as the differences in performance are magnified and more easily recognized.

We appreciate the comments made by Mr. Huebner. I might add, that I also would like to see this work correlated with the analytical approach being made at the Case Institute of Technology.

We also thank Mr. Ashley for his comments and we will certainly take into account the suggestions regarding these investigations.


1505

No. 1505

COOLING A SMALL RESIDENCE USING A PERIMETER-LOOP DUCT SYSTEM†

By D. R. Bahnfleth,* C. F. Chen,** and H. T. Gilkey***

PREVIOUS INVESTIGATIONS[1-9] in summer cooling were conducted in Warm Air Heating Research Residence No. 1 during the summers of 1932 through 1938, and 1940. During the summer of 1952 a series of summer cooling investigations[10] was begun in Warm Air Heating Research Residence No. 2. Both of these residences were of tight frame construction and had basements. Because of the increasing use of year 'round air conditioning in homes equipped with perimeter heating systems, it seemed advisable to begin summer cooling studies in Research Residence No. 3, which is a low cost, basementless residence with a concrete slab floor.

The principal objectives of this investigation were:

1. Determination of the cooling load and its hourly variation when cooling Research Residence No. 3 under both day and night conditions.
2. Determination of the time lags in heat flow through the walls and ceiling.
3. Comparison of the actual cooling load with the cooling load calculated according to the procedure outlined in The Guide 1952.[11]
4. Determination of the characteristics of the warm-air perimeter-loop system when used as a distribution system for summer air conditioning.
5. Determination of the operating characteristics of the mechanical condensing unit installed in the Residence.

Description of Residence

Residence No. 3 is a single-story, low cost home with a concrete slab floor. It is of standard frame construction, and has a relatively large amount of glass area. The south exposure is shown in Fig. 1. The walls are uninsulated, but the ceiling is insulated with 3⅝ in. thick bat-type mineral wool insulation. The exposed wall section consists of a double course of cedar shingles, building paper, shiplap sheathing on 2-in. × 4-in. studs, ½-in. gypsum board with aluminum foil backing and a simulated plaster finish. The attic is vented by louvered openings in the gable ends. The calculated coefficients of heat transmission U for the walls and ceiling are 0.21 and 0.07 Btu per (hr) (sq ft), respectively.

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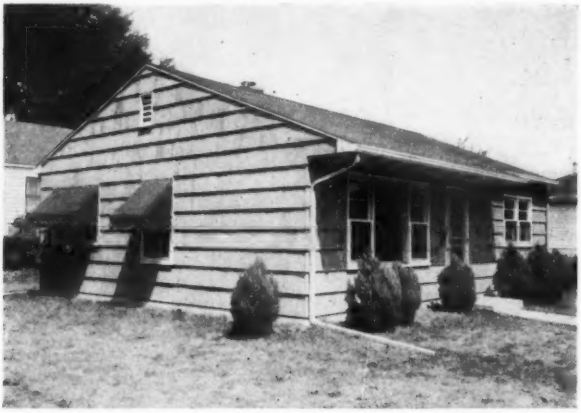


FIG. 1. SOUTHWEST VIEW OF RESEARCH RESIDENCE NO. 3

Except for one fixed picture window, well-fitted double-hung wood-sash windows are used throughout. The windows are not weatherstripped nor are they equipped with storm sash, but those on the east and west exposures are shaded by canvas awnings. The doors are of conventional wood and glass construction.

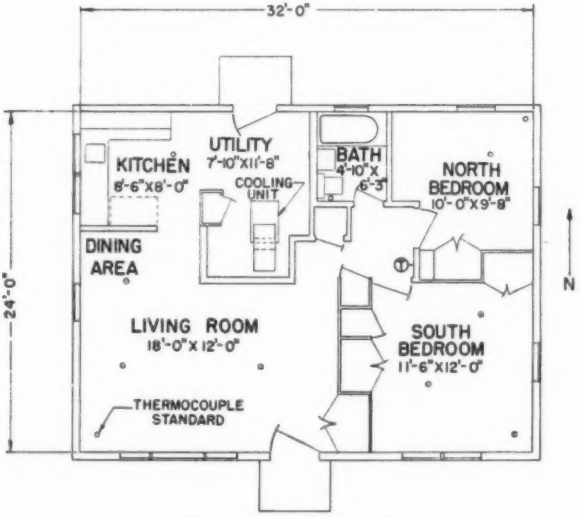


FIG. 2. FLOOR PLAN OF RESEARCH RESIDENCE NO. 3

The south exposure of the living room is shaded by a 3-ft roof overhang; the remainder of the south wall and the north wall are shaded by 1-ft roof overhangs.

The floor-slab construction consists of a 4-in. fill of coarse graded gravel placed on the original grade, a heavy duplex-paper vapor barrier (dampproofing membrane), and 4 in. of concrete. The insulation at the edge of the slab

TABLE 1—DATA FOR RESEARCH RESIDENCE NO. 3

A. Heat Transmission Coefficients, Btu per hr (sq ft) (F deg)	*U*
Uninsulated Frame Wall	0.21
Insulated Ceiling, with 3⅝ in. mineral wool insulation	0.07
B. Infiltration Factors, cu ft per hr (ft of crack)	*I*
Doors, no storm door	69
Windows, no weatherstripping, no storm sash	21
Fixed Window in Living Room	8

C. Room Dimensions,[a] Ceiling Area, Volume and Glass Area

Room	Dimensions		Ceiling Area Sq Ft	Exposure	Net Wall Area Sq Ft	Glass Area Sq Ft	Volume Cu Ft
	Ft In.	Ft In.					
Living Room	18 2	x 12 0	251	West	117	11	2,008
	8 6	x 4 0		South	87	58[b]	
Closet	4 0	x 2 4	9	South	19	0	72
South Bedroom	12 0	x 11 6	138	East	87	9	1,104
				South	73	19	
Closet	6 0	x 2 4	14				112
Closet	4 0	x 2 4	9	East	19		72
North Bedroom	9 8	x 8 0	92	North	71	9	736
	6 3	x 2 3		East	68	9	
Closet	4 0	x 2 4	9				72
Bath	6 3	x 5 5	34	North	39	6	272
Utility	12 0	x 8 0	96	North	46	18[b]	768
Kitchen	8 4	x 8 0	67	North	65		536
				West	45	19	
Hall	7 8	x 5 8	44				352
Closet	2 4	x 2 0	5				40
Totals			768		736	158	6,144

[a] Ceiling heights of first story—8 ft.
[b] Includes area of outside door which is of light wood panel and glass construction.

consists of 1-in. asphalt-sheathed glass fiber insulating board placed against the footing wall and extending downward 12 in. from the top of the slab.

A floor plan of the residence is shown in Fig. 2. The inside dimensions are 24 ft × 32 ft, and the corresponding floor area is 768 sq ft.

The cooling load of the structure, except for the windows, was calculated[11] by the sol-air temperature method. The heat gain through the windows was calculated using the tables recommended in THE GUIDE. The infiltration load was based on a wind velocity of 10 mph and the actual linear feet of crack around the doors and windows. Table 1 gives a summary of the room dimensions and volumes, and a compilation of the calculated cooling load for each

room is given in Table 2. The cooling load calculations were based on outdoor design conditions of 95 F dry bulb and 76 F wet bulb, and indoor conditions of 75 F dry bulb and 62.5 F wet bulb (50 percent relative humidity). The maximum calculated cooling load on a design day was 19,522 Btu per hr at 3 p.m. CST. The design heat loss of the residence for an outdoor temperature of —10 F and an indoor temperature of 70 F was about 51,600 Btu per hr when using

TABLE 2—COMPILATION OF CALCULATED COOLING LOAD FOR RESEARCH RESIDENCE NO. 3

A. Maximum Load (Conducted and Transmitted) at 3 p.m. CST, Btu per hr

ROOM	WINDOWS	EXTERIOR WALLS	CEILING	TOTAL
Living Room	2,910	1,630	1,040	5,580
Closet	[a]	[a]	[a]	[a]
South Bedroom	1,452	1,041	640	3,133
Closets (2)	[a]	[a]	[a]	[a]
North Bedroom	567	616	593	1,776
Closet	[a]	[a]	[a]	[a]
Bath	222	173	135	530
Kitchen	1,520	597	266	2,383
Utility	665	203	382	1,250
Hall	[a]	[a]	[a]	[a]
Closet	[a]	[a]	[a]	[a]
Totals	7,336	4,260	3,056	14,652

B. Grand Total Load, Btu per hr

	SENSIBLE	LATENT	TOTAL
Heat gain to rooms	14,652	—	14,652
Infiltration load	1,125	1,445	2,570
Load of blower (600 cfm)	525[b]	—	525
Total for system			17,747
10 percent miscellaneous gains			1,775
Grand total			19,522

[a] Cooling load for these rooms included with larger adjoining rooms.
[b] Measured load.

the perimeter-loop duct system. The residence was furnished but unoccupied during the studies.

EQUIPMENT AND INSTRUMENTATION

Cooling Unit: The cooling unit was a separate unit installed in the utility room of the residence and occupied a floor area of 22 in. × 36 in. For the purposes of this investigation the counterflow gas-fired furnace used during previous heating investigations[12] was removed from the residence. The conditioned air was circulated by the forward-curved blade centrifugal fan (10-in. diam, ¾ length wheel), hereafter called a blower, which was mounted integrally in the cooling unit, and was driven by a ½ hp motor. The air was drawn through the return-air duct into the cooling unit, through a filter, and then

through the evaporator and to the blower by which it was forced through a downcomer into the subfloor plenum. The filter, cooling coil, and the blower were located in an insulated compartment and were thus isolated from the condensing unit in the lower half of the housing.

The rated capacity of the conditioner was 24,000 Btu per hr under *A.S.R.E.* Standard inlet air conditions of 80 F dry bulb and 67 F wet bulb entering the cooling coil and 95 F water leaving the condenser. The semi-hermetic compressor was directly connected to a 220 volt single phase 2 hp motor. Both the compressor motor and the double-tube condenser were water cooled. The refrigerant used was dichlorodifluoromethane (F-12).

Duct System: The perimeter-loop system used during the investigation was the same as that used during previous[12] heating investigations. The loop system[13] consisted of a 6-in. diameter duct embedded in the concrete slab in the form of a single loop around the periphery of the floor and five 6-in. diameter

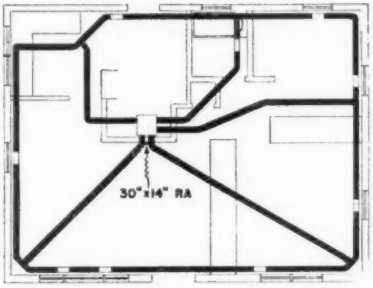


FIG. 3. PERIMETER-LOOP DUCT SYSTEM INSTALLED IN RESIDENCE

Floor registers — 4 in. × 14 in.
All ducts 6 in. diameter

feeder ducts connecting the perimeter loop to a subfloor plenum as shown in Fig. 3. The top of the feeder ducts was 6 in. below the floor surface at the plenum and sloped upward to the junction with the perimeter loop which was 2 in. below the floor. The air was delivered to the rooms through nine 4-in. × 14-in. straight-vaned floor registers located under the windows at the outside wall. The air was returned to the unit through one return-air grille (14 in. × 30 in.) located high in the north wall of the living room.

Instrumentation: There were 303 copper constantan thermocouples installed in the residence, 208 of 24-gage wire and the remaining 95 of 34-gage wire. Air temperatures were measured at four levels, at positions indicated by small circles on Fig. 2. Floor-surface temperatures were measured at 118 points. In addition, temperatures were measured in the attic, in the ground under the slab, in the cooling unit and the air distribution system, on the interior and exterior surfaces of the residence, and in the outdoor air. Wet bulb temperatures were measured by thermocouples made of 24-gage constantan and 36-gage copper wire. All of the thermocouples installed were connected to an indicating potentiometer by a system of rotary selector switches. A continuous record of 18

temperatures could be made by means of two recording potentiometers. The temperature measuring instruments were located in a garage adjacent to the residence.

Five heat-flow meters were installed at various locations in the residence. The meters installed in the walls were embedded in the gypsum wall board 60 in. above the floor, and were covered with a thin coat of plaster. The plaster was then painted to conform to the color of the room. The heat-flow meter installed in the ceiling was located between the room side of the insulation and the attic side of the gypsum ceiling board.

Continuous records of solar intensity, outdoor-air dry and wet bulb temperatures, and wind velocity were made. The wind direction was recorded at the times when temperature observations were made.

EXPERIMENTAL CONDITIONS

Four series of studies were conducted in the residence during the cooling season. The series designations and controlled variables are as follows:

SERIES	DATE	CONTROLLED VARIABLES
A-1	June 24 to July 19	Blower operated continuously; capacity approximately one-half of rating.
B-1	July 25 to Aug. 6	Blower cycled with compressor, capacity approximately one-half of rating.
A-2	Aug. 21 to Aug. 31	Blower operated continuously; capacity approximately two-thirds of rating.
B-2	Aug. 7 to Aug. 20	Blower cycled with compressor; capacity approximately two-thirds of rating.

The air-flow rate through the cooling unit was approximately 600 cfm, or 300 cfm per rated ton of refrigeration. No ventilation air was introduced mechanically into the residence. The thermostat was set to maintain a temperature of 75 F at the 30-in. level in the house. No attempt was made to control humidity.

Because the draperies were open, the only shading of the windows of the residence was provided by the previously mentioned awnings and roof overhang. The doors and windows remained closed throughout all studies. The residence was completely furnished but unoccupied.

TEST PROCEDURE

Temperatures of room air, outdoor air, floor surface, and interior and exterior wall surfaces were recorded daily at 3:00 p.m. CST, near the time that the peak load occurred. Power consumption, operation time and water consumption of the cooling unit were recorded each day at 5:30 a.m. CST, the time selected to start the 24-hr test day.

Special studies of from 24 to 72 hr duration were made periodically for all four test series. These studies were made during periods of clear hot weather to determine: (1) Time at which the maximum load was imposed on the residence and, in turn, on the cooling unit; (2) Time lag of the heat flow through the walls; (3) Effect of the length of unit operating cycles on the floor-surface temperatures; (4) Effect of continued hot weather on the cooling

load. Temperature and heat-flow observations were recorded hourly for approximately 18 hr of each day during the special-study periods.

DETERMINATION OF ACTUAL COOLING LOAD

The actual cooling load of the residence was determined by a heat balance which consisted of three components: (1) Heat removed from the conditioned air in the evaporator, (2) Heat absorbed by the condenser cooling water, and (3) Heat equivalent of the power input to the compressor.

Since a detailed explanation of the procedure of the heat balance was published in a previous paper,[10] only a brief description is presented in the following discussion.

It was assumed that heat absorbed by the cooling water was the sum of the heat absorbed by the refrigerant in the evaporator and the heat equivalent of the power input to the compressor. The condensing unit housing was completely insulated and the compressor motor was cooled by the condenser cooling water. The actual cooling load could then be found by subtracting the heat equivalent of the power input to the compressor from the heat absorbed by the condenser cooling water. The former could be evaluated by converting the power consumed by the compressor into its heat equivalent, while the latter could be found by knowing the temperature rise and the flow rate of the cooling water.

The actual cooling load was also calculated from the sensible and latent heat removed from the conditioned air. The sensible load was calculated from the air-flow rate and the air-temperature drop through the evaporator coils, and the latent load was found by measuring the quantity of water condensed from the conditioned air. The differences between the cooling loads as evaluated by these two methods for Series A-1, B-1, and B-2 were found to be within 4 percent. This difference could not be determined for Series A-2 because the continuous blower operation and frequent compressor operations resulted in little or no condensate being drained from the cooling unit. Typical operating data representing all test series are shown in Table 3. These data were taken from the daily 3:00 p.m. CST observations.

Some difficulty would be experienced in comparing the measured condensation rate with that calculated using the entering and leaving air conditions and the air-flow rate. The discrepancies result from such factors as moisture storage on the coil, the time element involved in the readings, and errors in measuring the wet-bulb temperatures. The measured condensation rate was based on the moisture collected during a one-hr time interval; the temperatures tabulated were instantaneous readings taken at the beginning of the compressor cycle.

TIME LAGS

The time lags in heat flow between the outer and inner surfaces of the walls and ceiling of a residence are important in that they may determine the time of maximum cooling load. Time lag is defined as the interval between the times of occurrence of the exterior and interior maximum surface temperatures, assuming that constant conditions are maintained within the conditioned space. The time lags, as thus defined, are affected by many variables besides the thermal properties of the materials used in the construction of a wall. Such factors as the direct solar radiation through the windows will affect the interior surface temperatures and consequently the apparent time lags. The maximum values of

TABLE 3—TYPICAL OPERATION DATA AND RESULTS AT 3:00 P.M. CST

SERIES[a] DATE	UNIT	A-1 JUNE 30	A-2 AUGUST 25	B-1 AUGUST 3	B-2 AUGUST 14
1. Length of time unit had operated continuously before readings	hr	6.67	0.22	6.42	0.23
2. Outdoor Air Conditions					
Dry Bulb	F	94.8	92.0	94.7	90.0
Wet Bulb	F	80.4	68.4	76.9	71.7
Relative Humidity	%	54.0	30.0	45.0	42.0
Dew Point	F	75.7	55.9	70.2	63.3
Humidity Ratio[b]		0.01925	0.00950	0.01590	0.01248
Wind. (Direction)—(Velocity, mph)		S-4	S-4	WSW-6	W-6
3. Indoor Air Conditions					
Dry Bulb	F	76.5	75.1	75.9	75.0
Wet Bulb	F	67.0	62.6	64.9	62.3
Relative Humidity	%	61.5	53.0	56.0	49.0
Dew Point	F	62.3	55.2	59.0	54.7
Humidity Ratio[b]		0.01204	0.00927	0.01068	0.00910
4. Condition of Air Entering Coil					
Dry Bulb	F	77.7	75.7	77.0	75.8
Wet Bulb	F	67.0	64.1	63.5	60.4
Relative Humidity	%	58.0	53.0	47.0	41.0
Dew Point	F	61.7	57.6	55.7	50.3
Humidity Ratio[b]		0.01175	0.01013	0.00947	0.00771
5. Condition of Air Leaving Coil					
Dry Bulb	F	61.5	55.4	59.0	50.5
Wet Bulb	F	59.2	53.9	57.0	48.5
Relative Humidity	%	88.0	90.0	88.0	86.0
Dew Point	F	57.8	52.8	55.6	46.6
Humidity Ratio[b]		0.01020	0.00848	0.00944	0.00673
6. Air Temperature Drop Through Coil	F deg	16.2	20.3	18.0	25.3
7. Temperature of Cooled Air Leaving Registers (Average of all)	F	67.0	65.0	65.4	63.0[c]
8. Air Temperature Rise,					
Unit to Subfloor Plenum	F deg	1.3	4.2	2.7	4.1[c]
In Feeders		2.2	3.0	1.8	3.9[c]
In Loop		2.0	2.2	1.9	3.5[c]
9. Floor-Surface Temperature (Average)	F	73.4	70.5	72.6	71.3
10. Flow Rate of Recirculated Air	cfm	596	598	605	610
11. Moisture Condensed from Air	lb/hr	0.69	[d]	0.8	1.53
12. Heat Absorbed by Cooling Coil,					
Total	Btu per hr	10,910	[d]	12,350	17,870
Latent		750	[d]	880	1,650
Sensible		10,160	12,800	11,470	16,220
13. Water Temperature	F				
Entering Condenser		58.3	58.5	60.5	58.5
Leaving Condenser		71.0	79.6	96.0	79.7
Rise		12.7	21.1	35.5	21.2
14. Flow Rate of Condenser Water	gpm	2.72	2.15	1.06	2.18
	lb/hr	1360	1075	532	1090
15. Heat Absorbed by Water in Condenser	Btu/hr	17,270	22,700	18,850	23,100
16. Heat Equivalent of Power Input to Compressor	Btu/hr	6,960	6,820	6,960	7,030
17. Net Cooling Load (Item 15—Item 16)	Btu/hr	10,310	15,880	11,890	16,070

[a] Series A-1 and A-2, blower operated continuously; Series B-1 and B-2, blower cycled with compressor.

[b] Pounds water per pound dry air.

[c] Data taken from August 15, 1953. Indoor conditions were similar to those for August 14, 1953.

[d] No condensate collected with continuous blower operation and with unit cycling frequently.

surface temperature and heat flow and their times of occurrence for each exposure of the residence are shown in Table 4. The last column of this table shows the apparent time lags for each exposure of the residence. The values were obtained from curves in which hourly readings of surface temperatures and heat flow had been plotted.

In Table 4 it can be seen that the observed time lags varied from day to day and from wall to wall throughout the cooling season. This inconsistent variation can be attributed to unsteady weather conditions and to the orientations of the walls. The exterior surface temperatures are affected by changes in wind velocity and direction, solar intensity, and outdoor temperature. Variation in time lags occurred because the interior surface temperatures are neither immediately nor directly affected by changes in the exterior-surface temperatures. The time lag for any given exposed wall section will also be affected by the orientations of the other exposed walls in the room, and by direct and diffuse solar radiation through the windows.

The radiation received from the other exposed wall sections and the windows can cause the interior surface temperature to reach a maximum at different times depending upon the time of peak load on the other wall sections. Thus, the time lag for an east wall of a residence should be relatively consistent since it will not be affected appreciably by the heat gains through the other exposed surfaces. That this is true is shown in Table 4. The time lags measured for the east wall during the four test series were 2.0, 0.9, 1.8, and 2.1 hr. The time lag of 0.9 hr, measured on August 25, was the result of the late time at which the maximum exterior surface temperature apparently occurred. Had the peak occurred at 9:00 a.m. as it did in the other two studies in August, the measured lag for the east wall on this day would have been 1.9 hr. The measured time lag variations for the other exposures were greater because of the effect of the heat gains through the other exposed surfaces from which they received radiation.

An average time lag for each exposed wall and the ceiling can be determined from Table 4 for the four days listed. These averages for the east, south, west, and north exposures were 1.7, 1.4, 1.2, and 1.3 hr, respectively. The average time lag for the ceiling was 2.1 hr for the four days. Throughout the investigations, the temperatures of the exterior and interior surfaces of the west wall of the residence were recorded continuously. The time lag for the west wall of the residence could then be determined for each day on which significant data were recorded. The average time lags for the west wall determined for 14 days during Series A-1 and 13 days during Series B-1, were 1.0 and 1.1 hr, respectively. These averages are in fair agreement with the average of the four days previously mentioned.

COMPARISON OF CALCULATED AND MEASURED HEAT FLOWS

Figs. 4, 5, and 6 show comparisons between the measured and calculated heat flows for two walls and the ceiling of the residence. Because the overall coefficient of heat transfer for the stud space of an uninsulated wall section varies only slightly from that of the entire uninsulated wall, no adjustments in the calculated heat-flow rates were made for the comparisons of heat flow through the walls. In the case of the ceiling, the heat-flow meter measured the heat flow through an insulated section of the ceiling. Thus, it was necessary to correct the calculated heat-flow rates through the ceiling so that they were based

TABLE 4—SUMMARY OF SURFACE TEMPERATURES, HEAT FLOWS AND TIME LAGS

SERIES	DATE	ORIENTATION	MAXIMUM VALUE AND TIME OF OCCURRENCE						TIME LAG[b] EXT TO INT SURF, HR
			Exterior Surf Temp		Heat Flow		Interior Surf Temp		
			F	CST	Btu/hr (sq ft)	CST	F	CST	
A-1	June 30	East	146.8	8:00 a.m.	11.50	9:12 a.m.	87.5	10:00 a.m.	2.0
		South	100.5	2:00 p.m.	4.18	1:24 a.m.	83.8	1:30 p.m.	−0.5
		West	125.0	3:00 p.m.	7.75	5:00 p.m.	84.2	5:00 p.m.	2.0
		North	100.0	2:00 p.m.	3.25	4:24 p.m.	80.7	3:00 p.m.	1.0
		Ceiling[a]	117.8	2:30 p.m.	4.07	2:00 p.m.	82.5	5:00 p.m.	2.5
A-2	August 25	East	129.0	10:00 a.m.	8.27	10:54 a.m.	84.0	10:54 a.m.	0.9
		South	135.7	11:54 a.m.	7.83	1:00 p.m.	85.1	1:00 p.m.	1.1
		West	130.7	3:00 p.m.	5.30	4:00 p.m.	82.0	4:00 p.m.	1.0
		North	101.0	11:54 a.m.	3.27	1:00 p.m.	78.8	2:00 p.m.	2.1
		Ceiling[a]	128.2	1:00 p.m.	3.54	1:00 p.m.	80.3	2:00 p.m.	1.0
B-1	August 3	East	140.5	9:06 a.m.	10.87	10:00 a.m.	86.4	10:54 a.m.	1.8
		South	130.3	12:45 p.m.	6.40	2:00 p.m.	85.2	1:18 p.m.	0.6
		West	137.2	3:54 p.m.	7.80	5:00 p.m.	85.0	4:36 p.m.	0.7
		North	103.2	1:00 p.m.	3.57	5:00 p.m.	80.7	2:00 p.m.	1.0
		Ceiling[a]	132.0	1:30 p.m.	3.97	2:00 p.m.	81.8	4:12 p.m.	2.7
B-2	August 14	East	140.8	8:54 a.m.	10.33	9:54 a.m.	85.9	11:00 a.m.	2.1
		South	129.7	12:00 noon	6.48	1:00 p.m.	84.5	1:36 p.m.	1.6
		West	131.5	3:00 p.m.	6.60	4:00 p.m.	84.0	4:00 p.m.	1.0
		North	95.2	2:00 p.m.	2.72	5:00 p.m.	79.2	3:00 p.m.	1.0
		Ceiling[a]	129.0	2:00 p.m.	3.72	2:30 p.m.	80.7	4:00 p.m.	2.0

[a] The exterior surface of the ceiling was considered to be the attic side of the insulation.
[b] Predicted time lags for all walls — 2.0 hr.
Predicted time lags for roof and ceiling — 3.0 hr.

on an overall coefficient of heat transmission U of 0.06 Btu per (hr) (sq ft) rather than the value of 0.07 Btu per (hr) (sq ft) used for the cooling load calculations.

In a previous investigation[10] it was shown that, in general, the measured heat-flow rate was less than the calculated rate when the wall was not exposed to direct solar radiation. Further, it was shown that when a wall was exposed

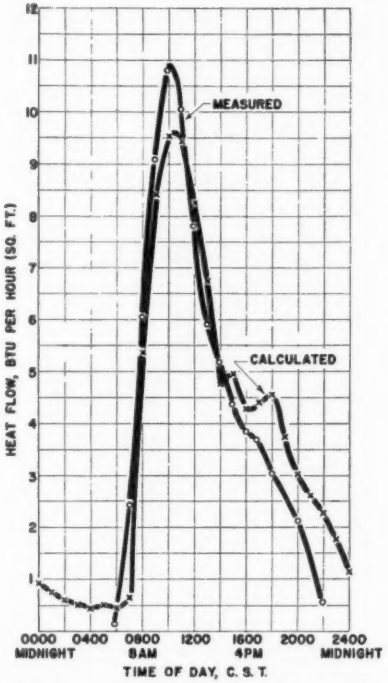


FIG. 4. CALCULATED AND MEASURED HEAT FLOW THROUGH EAST WALL FOR AUGUST 3, 1953

to direct solar radiation, the measured value was greater than the calculated value. These same general conclusions can be drawn from Fig. 4 and from similar studies made on the north and west walls of the residence. In Fig. 4 it can be seen that in the morning from 6 a.m. to 11:30 a.m. when the east wall received direct solar radiation, the measured heat flow was greater than that calculated. Throughout the rest of the day, while the wall was in the shade, the measured heat flow was less than that calculated by the sol-air temperature method. The maximum measured heat flow of 10.87 Btu per (hr) (sq ft) was 11.7 percent greater than the maximum calculated heat flow of 9.60 Btu per (hr) (sq ft) for the uninsulated east wall. In the case of a fully insu-

lated west wall exposed to direct solar radiation, it has been shown[10] that the maximum measured heat flow was 36 percent greater than the maximum calculated heat flow.

The roof-attic-ceiling combination was treated as a flat roof when calculating the cooling load. Recognizing that the sol-air temperatures for horizontal surfaces did not apply directly to such a combination, these sol-air temperatures were reduced by 22 percent. As shown in Fig. 5, this assumption resulted in a calculated heat flow less than the measured heat flow. This would be more evident if the calculated heat flow curve were shifted to correct for the difference in the predicted and measured time lags. The maximum measured heat flow of 3.96 Btu per (hr) (sq ft) was 5 percent greater than the maximum calculated heat flow of 3.75 Btu per (hr) (sq ft). In a previous investigation,[10]

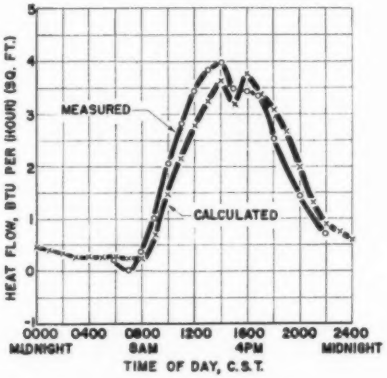


FIG. 5. CALCULATED AND MEASURED HEAT FLOW THROUGH CEILING FOR AUGUST 3, 1953

when the same assumptions were made in calculating the heat flow through the ceiling, the maximum measured heat flow of 2.1 Btu per (hr) (sq ft) was 57 percent less than the maximum calculated heat flow of 3.3 Btu per (hr) (sq ft). The marked difference in the two results was due in part to the different amounts of free ventilation in the attics of the two residences.

The measured heat flow was always less than the calculated heat flow through the south wall of the residence (see Fig. 6). The calculated heat flow was that for an unshaded south wall. It should be noted in Table 4, however, that the heat flow through the south wall increased as the season progressed. This increase in heat flow was due to the increased time of exposure of the heat-flow meter location to direct solar radiation. During the early part of the season, this heat-flow meter station was shaded completely by the 1-ft roof overhang. Near the end of the cooling season this roof overhang provided shade only during the mid-morning, and in the afternoon it was shaded by the 3-ft roof overhang on the south wall of the living room. The differences between the measured and calculated heat flows for June 30, August 3, August 14 and

August 25 were 61.5, 48.8, 7.3, and 2.9 percent, respectively. If the south wall had been completely unshaded, it is doubtful if the calculated heat flow would have been greater than the measured heat flow.

COMPARISON OF ACTUAL AND CALCULATED COOLING LOADS

The 1952 edition of THE GUIDE was used for all cooling load calculations because the cooling load could be determined for each hour of the day by the sol-air temperature method. The wall transmission loads for the days selected

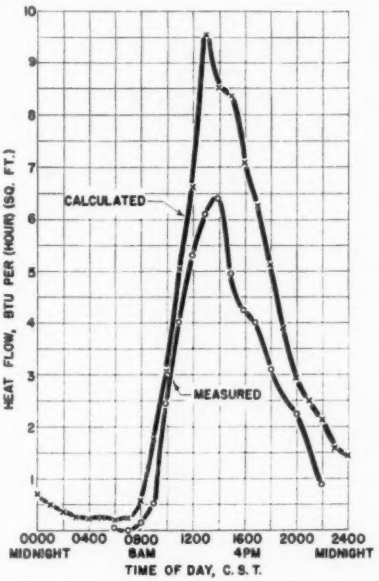


FIG. 6. CALCULATED AND MEASURED HEAT FLOW THROUGH SOUTH WALL FOR AUGUST 3, 1953

for the comparison of actual and calculated cooling loads were calculated by the sol-air temperature method. The design sol-air temperatures were adjusted for the outdoor-air temperatures experienced in accordance with the correction factors given in THE GUIDE 1952. It was assumed that the indoor design conditions were maintained. Because the design sol-air temperatures did not directly apply to the combination of gable roof, vented attic, and insulated ceiling, the surface absorptivity for solar radiation on horizontal surfaces was assumed to be 0.7 rather than the value of 0.9 suggested in THE GUIDE 1952, for dark roof surfaces. This reduction in absorptivity lowered the design sol-air temperatures approximately 22 percent.

The south wall was considered to receive direct solar radiation even though a large portion of it was shaded by the roof overhang. Although it is recom-

mended[11] that sol-air temperatures for a north wall be applied to all shaded walls, differences in the values of sky radiation and in times of peak temperatures indicated this would introduce an error. The error introduced by considering the south wall unshaded would be on the conservative side.

The heat gain through the windows was computed from tables presented in THE GUIDE 1952. A shading factor of 0.3 was applied to direct solar radiation

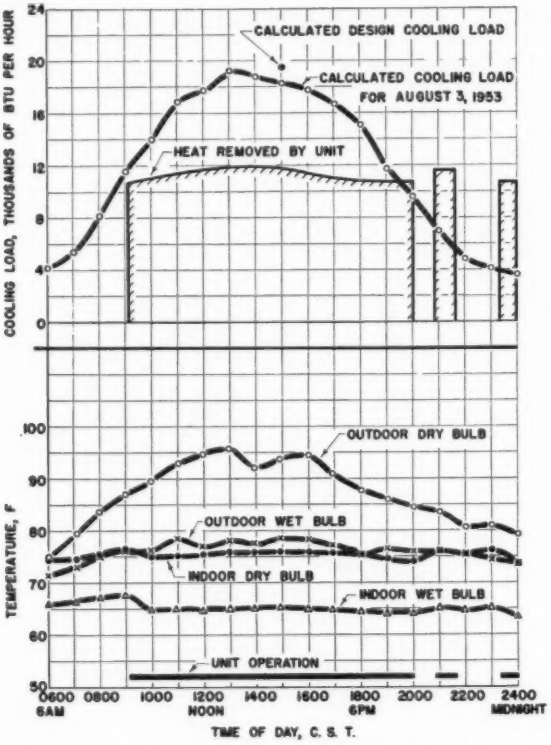


FIG. 7. ACTUAL AND CALCULATED COOLING LOADS OF RESIDENCE FOR AUGUST 3, 1953 (SERIES B-1)

gains for all windows equipped with awnings or shaded by the roof overhang. The heat gains through the glass by conduction and convection were corrected for the outdoor-air temperatures experienced in accordance with the recommended correction factors.

Even though outdoor design conditions were exceeded many times during the investigations, the cooling unit never operated with a capacity greater than 16,000 Btu per hr, and indoor design conditions were maintained. With a cooling unit capacity of approximately 12,000 Btu per hr and on a design day, the

indoor design conditions of 75 F dry bulb and 62.5 F wet bulb were only slightly exceeded. Fig. 7 shows the operating times and cooling load on August 3, 1953 (Series B-1). As mentioned previously, the calculated cooling load for this day was adjusted for the outdoor-air temperatures experienced, and the actual cooling load was obtained from a heat balance. Similar data were obtained for the other three series.

In all cases, the total calculated heat gain for the period 6:00 a.m. to midnight exceeded the total heat removed by the cooling unit during the same period by at least 57 percent. For Series B-1 tests the total calculated heat gain was 220,880 Btu during the 18-hr period, while the measured heat removed was 140,300 Btu. This was a difference of 57.4 percent based on the measured value. For the three other series, A-1, A-2, and B-2, the differences in total heat gain

TABLE 5—COMPARISON OF MEASURED AND CALCULATED HEAT REMOVAL BY UNIT

PERIOD	SERIES A-1, JUNE 30, 1953			SERIES B-1, AUGUST 3, 1953		
	Measured	Calculated	Percent Difference Based on Measured	Measured	Calculated	Percent Difference Based on Measured
6:00 — 8:00 a.m.	0	14,100	—	0	11,200	—
8:00 — 10:00 a.m.	14,100	26,000	84.4	10,100	22,400	122.0
10:00 — 12:00 noon	22,700	33,900	49.3	22,400	32,800	46.4
12:00 — 2:00 p.m.	21,500	37,900	76.2	23,900	37,200	55.6
2:00 — 4:00 p.m.	20,900	38,200	82.7	23,900	36,500	52.7
4:00 — 6:00 p.m.	21,500	35,200	64.7	22,400	34,000	51.8
6:00 — 8:00 p.m.	21,400	26,500	23.8	21,600	24,300	12.5
8:00 — 10:00 p.m.	6,560	16,100	144.0	9,280	14,400	55.2
10:00 — 12:00 mid.	0	9,760	—	6,720	8,080	20.2
Totals.....	128,660	237,660	+84.9	140,300	220,880	57.4

based on the measured value were 84.9, 104.0 and 107.0 percent, respectively. The large differences in the latter two series (A-2 and B-2) can be attributed in part to the low outdoor-air temperatures experienced during the night hours and the consequent flywheel effect of the residence and the concrete slab floor. The differences for the three series, A-1, A-2, and B-2, can be attributed partly to deviations of actual solar conditions from those used as a basis for the design sol-air temperatures.

The calculated and actual cooling loads for nine 2-hr periods on June 30 (Series A-1) and on August 3 (Series B-1) are shown in Table 5. The values were determined by integrating the areas under the curves of measured and calculated cooling loads. The data in Table 5 show the typical trends for the two methods of blower operation investigated. The differences between the measured and calculated cooling loads are large for most of the 2-hr periods during the day. During the period of the first compressor operations of the day, a small difference in the loads for Series A-1 (blower operated continuously) was observed as compared with a large difference observed for Series B-1 (blower cycled with compressor). Near the end of the compressor operations for the day, a large difference between the measured and calculated cooling loads was observed for Series A-1 as compared to a small difference

observed for Series B-1. The large differences between the measured and calculated cooling loads which occurred at the beginning and end of daily compressor operation for the two series resulted from the type of blower operation used.

When the blower operated continuously, Series A-1, the circulating air reduced the night-time fly-wheel effect of the slab floor. In the day time, the internal energy of the concrete slab decreased during the operation of the compressor. When the compressor ceased to operate, the circulating air increased the internal energy of the slab until an isothermal condition was approached in the residence. The heat gains that occurred during the late evening hours were absorbed by the slab in this manner, and therefore did not cause further compressor operation. Since the cooling unit did not have to operate to maintain the control temperature within the residence, a large difference in measured and calculated cooling loads resulted. The heat gains which occurred in the morning were not absorbed by the slab because of the small temperature differences between the slab and the room air and surrounding surfaces. Since these heat gains were not absorbed by the slab, they raised the room-air temperature above the control point, causing the compressor to operate. When the air temperature within the residence responded directly to the heat gains, the difference in actual and calculated cooling loads was small.

When the blower cycled with the compressor, the internal energy of the concrete slab floor was raised only slightly during the late evening hours. For this reason, the heat gains raised the room-air temperatures causing the compressor to operate. The response of the room-air temperature to the heat gains caused the small differences observed in the actual and calculated cooling loads. With the blower cycling with the compressor, the internal energy of the concrete slab was not raised appreciably during the night; hence, lower floor-surface temperatures were observed in the morning. Due to the temperature differences between the slab and its surroundings when the heat gains began to increase in the morning, the slab absorbed these gains causing the apparently large difference between the measured and calculated cooling loads.

Although a concrete slab floor will act as a thermal flywheel, the effect of the slab on the actual cooling load is not known. The preceding discussion should not be interpreted as a statement of percentage effects that a concrete slab floor will have on the cooling load of a small residence. Rather, it serves to point out that a slab will have some leveling effect on the cooling load in that it will minimize the cyclic variation of the heat gain. Hence, a smaller cooling unit will be able to provide more comfortable conditions in a residence with a concrete slab floor than that predicted by the usual methods of calculating the cooling load.

FLOOR-SURFACE TEMPERATURES AND ROOM-AIR DEW POINTS

Floor-surface temperatures in the vicinities of the subfloor plenum, the feeder ducts, and the perimeter loop were found to vary with the length of compressor operation during the day. These temperatures decreased as the compressor operation time increased. When the cooling unit ceased operation late in the day, a marked increase of these floor surface temperatures was observed. This cyclic phenomenon was more noticeable in Series A-1 and A-2 than in Series B-1 and B-2, since the floor slab was cooled by the conditioned air when the compressor was on and heated by the recirculated room air when the compressor

was off. In both series in which the blower cycled with the compressor (B-1 and B-2), when the compressor was off, the air in the embedded ducts absorbed heat from its immediate surroundings and finally reached a thermal equilibrium so that a fairly constant temperature was maintained until the next compressor operation.

The minimum floor-surface temperatures were found near the subfloor plenum. Floor-surface temperatures in areas remote from the ducts and the subfloor plenum were found to equal or exceed the room-air temperature. This was attributed to diffuse radiation from the walls and the ceiling to the cooler floor surface. Floor-surface temperatures directly over the feeder ducts were found to be uniform from the pit to the loop. This was attributed to the upward slope of the feeder ducts from the pit to their junctions with the loop duct.

The room-air dew-point temperatures experienced in Series A-1 and A-2 were higher than those experienced in Series B-1 and B-2. When the blower operated continuously, water which had been condensed but not drained out of the evaporator coils during compressor operations was re-evaporated. The amount of re-evaporation was calculated by two methods for July 18 (Series A-1) after the cooling unit had operated continuously for 8 hr and 25 min. In one case the amount of re-evaporated moisture was determined by the conditions of the air entering and leaving the cooling unit. In the second method, room-air conditions served as the basis of calculation.

After the compressor was off for 1.67 hr, the amount of re-evaporated moisture as calculated by the first method was 2.24 lb, while a value of 1.69 lb was determined by the second method. The difference of 0.55 lb of water can be considered to be absorbed by the house and furnishings. Within the same period, the room-air dew point increased from 58.2 to 67.0 F.

During the testing period, from June 24 to August 31, there was no visible evidence of condensation on the floor surface. The daily maximum room-air dew point and the minimum floor-surface temperature at that same time were evaluated and are shown in Fig. 8. The fact that the maximum room-air dew point was always lower than the minimum floor-surface temperature further substantiated that condensation on the floor could not have occurred.

OPERATING CHARACTERISTICS

The air distribution in the living room was investigated during one long cycle of the cooling unit. When the unit was operating at the higher capacity and consequently on short running cycles, an isothermal condition in the living zone was approached during each off cycle. Since this would tend to eliminate any stratification or drafts caused by the delivery of conditioned air from floor registers, the long operation was considered to be the worst condition which would be encountered.

During the air distribution study, room-air temperatures and velocities were taken at four levels in the living room. At no point in the living zone did the room-air velocity exceed 32 fpm except immediately above the supply outlets. At the floor level, where the lowest room-air temperatures were experienced, the room-air velocities did not exceed 15 fpm. This low velocity was considered desirable because of the low air temperatures in this region. During the study the maximum observed temperature difference from the floor level to the ceiling

level was 2.7 deg. That the air distribution experienced when floor registers were used was acceptable for cooling was shown by this study and by the temperature conditions maintained in the residence throughout the cooling season.

The temperature variations at the 30-in. level observed in Series A-1 and B-1 were small. Due to the low capacity of the cooling unit in these two series, it usually operated continuously for a long period, thus keeping the room-air temperatures at fairly constant values. The difference between the maximum room-air temperature which occurred just prior to the compressor operation and the minimum room-air temperature which occurred at the end of long compressor operation was found to be less than 2 deg. This slight variation of room-air temperature is shown in Fig. 7. When the cooling unit was operating at the higher capacity (Series A-2 and B-2), frequent and short cycles were experienced. Due to this condition, the room-air temperature varied in a cyclic form about in step with the unit operation. The maximum variation was found to be 2.5 deg.

The room-air temperature balance was found to depend on the time of day and the type of operation. For instance, in the south bedroom, which was exposed on the east and south, higher room-air temperatures were experienced that in the rest of the rooms in the late morning just prior to the unit operation. A difference of 2 to 3 deg was found for Series A-1 and A-2, and a difference of 1 to 2 deg was found for Series B-1 and B-2 at that time of the day. The smaller difference in room-air temperatures between rooms experienced in Series B-1 and B-2 (blower cycled with compressor) was attributed to the cooling effect of the floor slab.

The room-to-room temperature balance at 3:00 p.m. CST for all four test series was found to be:

1. Series A-1: 0.5 to 1.8 deg
2. Series A-2: 1.0 to 1.8 deg
3. Series B-1: 0.4 to 1.4 deg
4. Series B-2: 0.6 to 1.4 deg

In both Series A-1 and B-1, maximum differences for most temperature measuring stations were less than 1.0 deg. This small room-to-room temperature difference was attributed to the low capacity of the unit and long unit operation in these series.

The cooling unit was operated at two refrigeration capacities during the studies. The lowest, which was observed when the installation of the cooling unit had been completed, was approximately 12,000 Btu per hr, half of its rated capacity. Two series of studies, A-1 and B-1, were made with the unit operating at this capacity. The capacity of the unit was raised to 16,000 Btu per hr for Series A-2 and B-2. The latter condition was still much below the rated capacity of the unit; however, this discrepancy was attributed to the differences between the rating conditions and those under which these studies were conducted. The high water consumption observed in Series A-1 was caused by an overcharge of refrigerant in the cooling unit. This overcharge, which caused the large water flow rate through the condenser, had no appreciable effect on the capacity of the cooling unit.

A summary of cooling unit operating characteristics is presented in Table 6. The compressor operated a total of 374 hr during the four test series. The

total power and water consumptions of the unit were 742 kw hr and 5,568 cu ft, respectively. The blower power consumption was fairly constant in all series with an average value of 0.15 kw. The total blower power consumption for the four series was 158 kw hr. If the blower had been operated continuously during the four studies, it would have consumed 232 kw hr. The blower would have used only 57 kw hr if it had been cycled with the compressor during the four series. Hence, a saving of 175 kw hr would have resulted if the blower had cycled with the compressor during the entire investigation.

The measured pressure losses in the air distribution system for an airflow rate of 600 cfm were 0.094 in. of water on the supply-air side of the system and

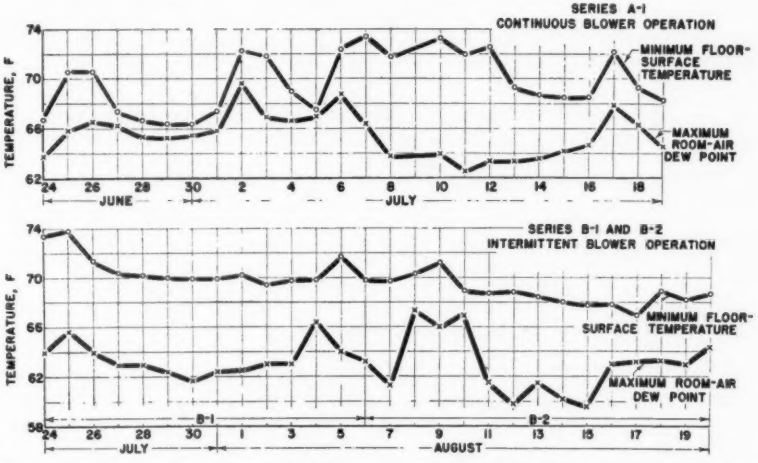


FIG. 8. DAILY MAXIMUM ROOM-AIR DEW POINT AND MINIMUM FLOOR-SURFACE TEMPERATURE

0.011 on the return-air side. This is a total of 0.105 in. of water for the system. For an airflow rate of 430 cfm, a total pressure loss of 0.130 in. of water was found with the same duct system during heating investigations conducted[12] in Research Residence No. 3. The difference in pressure losses experienced with heating and cooling was due to the increases in size of the return-air grille and duct and in the adjustment of the registers to obtain the most satisfactory room-to-room temperature balance when cooling the residence.

CONCLUSIONS

The time lag of the heat flow through the east wall of the residence varied only slightly during the studies, but the time lags for the other walls and the ceiling showed considerable variation. Average time lags determined from data for *design days* for the east, south, west and north walls were 1.7, 1.4, 1.2 and 1.3 hr, respectively. The average time lag for the ceiling was 2.1 hr for these days. The average time lag in heat flow through the west wall determined from recorded data for 27 days was approximately 1.1 hr.

In general, the measured heat-flow rates through walls exposed to direct solar radiation were higher than the calculated rates. The measured heat-flow rates through walls not exposed to direct solar radiation were less than the calculated rates. The 22 percent reduction in the sol-air temperatures for horizontal surfaces applied to the roof-attic-ceiling combination caused the measured heat-flow rate to be greater than the calculated rate. The difference between the measured and calculated heat-flow rates through the south wall decreased as the season progressed due to the increased time the heat-flow meter location was exposed to direct solar radiation. The calculated heat-flow rates were based on the sol-air temperatures for an unshaded south wall.

On a day when outdoor design conditions were exceeded, a room-air temperature rise of less than 1.5 F above control temperature was experienced even though the cooling unit was operating at half its rated capacity. In all studies, the total calculated heat gain for the period 6:00 a.m. to midnight exceeded the

TABLE 6—SUMMARY OF UNIT OPERATION CHARACTERISTICS

SERIES	DATE	MAXIMUM FOR ONE SERIES			PERIOD	TOTAL FOR SERIES			
		Hours Operation	Compressor Power, kwhr	Water Consumption cu ft		Hours Operation	Power Consumption, kwhr		Water Consumption cu ft
							Comp	Blower	
A-1	June 30	11.85	23.6	258.2[a]	June 24—July 19	170.57	347.7	94.3	2903.1[a]
A-2	August 31	8.30	17.2	148.6	August 21—August 31	55.88	97.1	40.4	1013.5
B-1	August 1	13.07	27.2	111.3	July 25—August 6	118.60	239.6	18.6	1093.6
B-2	August 14	4.63	9.5	89.8	August 7—August 20	28.52	58.0	4.6	558.1

[a] High water consumption due to an overcharge of refrigerant

total heat removed by the cooling unit during the same period by at least 57 percent. When the blower operated continuously, the largest differences between the measured and calculated cooling loads occurred near the end of compressor operation. With the blower cycled with the compressor, the largest differences in measured and calculated cooling loads occurred at the beginning of the compressor operation. The variance in times of occurrence of the large differences was attributed to the flywheel effect of the concrete slab floor.

During the testing period, June 24 to August 31, there was no visible evidence of condensation on the floor surface. Since the maximum room-air dew-point temperature was lower than the corresponding minimum floor-surface temperature at any time, condensation on the floor surface could not have occurred.

ACKNOWLEDGMENTS

This paper reports one phase of a comprehensive investigation conducted in Warm Air Heating Research Residence No. 3 under the terms of a cooperative agreement between the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station of the University of Illinois.

These results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

Acknowledgment is made to the manufacturers who cooperated by furnishing equipment used in the investigation.

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DISCUSSION

S. F. GILMAN, Syracuse, N. Y.: This paper represents another contribution made by the University of Illinois as the result of their research programs in the field of residential air conditioning.

Looking at Fig. 4, which shows the measured and calculated heat flows through the east wall, and realizing the walls are not insulated, it is clearly evident that insulation in at least east and west walls is very desirable.

The agreement between the calculated and measured heat flows seems very satisfactory for this east wall and also for the ceiling. It is not as satisfactory, however,

for the only other wall shown, the south wall. But here the authors assume the south wall to receive direct solar radiation, even though it is protected by an overhang, and give reasons for making the assumption.

I would like to ask if the assumption that the wall is completely shaded yields a maximum heat flow closer to the measured maximum.

If calculated and measured heat flows for walls and roof can be brought into agreement, further investigation by the authors might well yield an improved method of calculating heat gain through glass, such that the actual and predicted maximum of cooling loads would be much closer together. The benefits from being able to select the proper capacity equipment are self-evident.

The very complete data obtained from this and other research residences at Illinois might well be used in conjunction with the thermal circuit technique.

If and when the A.S.H.V.E. Research Laboratory continues studies on this subject, it is suggested that one of these residences be selected for analysis. The circuit technique solutions could then be compared with actual operative conditions.

It is interesting to see that the floor registers provided satisfactory air distribution. Herein lies an advantage of the panel convection type of system. In passing through the embedded ducts the temperature of the supply air rises such that the differential between the supply and room air temperatures is about 10 deg on the average, rather than the usual 15 to 20 deg for non-panel systems.

As shown by others, good air distribution under this condition is not as difficult a problem. I would, however, like to know the order of magnitude of the outlet air velocities and also if the authors plan to utilize flow diffusers rather than flow registers during any future studies.

L. T. AVERY, Cleveland, Ohio: I would like to ask what method was used to reduce the machine capacity and why. I am very much interested because in applying cooling to small installations it is of the utmost importance that the machine not cycle; and I rather imagine that's why the authors did it.

What supply air temperature was used? Mr. Gilman suggested it might have been 10 deg below the room temperature.

I also call your attention to the fact that we are talking now about 75 F room temperature, not 80 F. I think it is time we recognized in our literature that 80 F is a bit obsolete.

As a matter of interest, I might describe a perimeter-loop system in an office which we recently encountered. The office had plate glass windows exposed to the south and west with inside venetian blinds. The system had been laid out and was installed before we secured the order for the cooling and it was done entirely for the heating purpose, and we wondered what results we would get. They were so ideal that I would follow the design of that system if I were laying out a new installation. It is perfect from the cooling standpoint.

P. R. ACHENBACH, Washington, D. C.: I found this paper very interesting and it is not intended as any disparagement to say that it raised as many questions in my mind as it answered. I would like to present some of these questions to the authors.

It will be noted in Table 3 that the desired relative humidity level of 50 percent was not maintained during tests A-1 and B-1 when the compressor was operated at half capacity even though the compressor operated continuously. We ordinarily think continuous compressor operation promotes lower relative humidity in the living space. A somewhat lower relative humidity was maintained during cycling operation in tests A-2 and B-2 with the compressor operating at two-thirds capacity.

Although the authors were studying only the cooling load of the house itself, the sensible heat ratio would undoubtedly be lower if the house were occupied and if the air conditioning unit were used to bring in additional ventilating air. The infiltration allowed for in the computations amounts to about half an air change per hour. Thus,

a proportionately greater dehumidifying load would have been imposed on the air conditioner, if the house were occupied.

It is interesting to try to evaluate the latent heat load on the air conditioning unit during these tests. I can appreciate the difficulty of trying to obtain agreement between the amount of water drained from the evaporator coil and the change in humidity ratio of the air as it passes through the unit. The observed indoor and outdoor humidity ratio shown in Table 3 and the assumed infiltration rate indicate that about 1.2 lb of water would be condensed per hour. On the other hand, the rate of air circulation and change in humidity ratio between coil inlet and outlet for tests A-1 and A-2 suggests that approximately 4 lb of water was condensed per hour. However, the weighed condensate shows that less than one pound of water was condensed during the one hour period of collection. Thus the actual dehumidification load is somewhat uncertain for these tests.

The data in Table 3 show that from one-fourth to one-third of the total dry bulb temperature rise of the circulated air occurred in the feeder ducts and perimeter loop at 3 p.m. when the maximum computed cooling load occurred. This temperature rise would represent the combined effect of heat gained from the earth, heat gained from the outdoors through the perimeter loop, and heat gained through the concrete from the room above. The data do not reveal whether or not the heat capacity of the concrete slab levelled off the peak load on the cooling unit. A record of the variations on the upper and lower surface temperatures of the concrete slab throughout the day and night would indicate whether heat was alternately gained and lost by the slab as the inner surface temperatures of the walls and ceiling increased and decreased. A 6-in. concrete slab of the dimensions needed for the test house would absorb about 11,000 Btu for each degree F rise in average temperature of the slab.

I would also like to inquire whether the authors looked for condensation in the subfloor ducts. Fig. 8 shows that during one of the tests the minimum floor surface temperature was less than one degree warmer than the maximum dew point temperature. Data in Table 3 indicate that the air in the subfloor ducts was considerably lower than the floor surface temperature. It seems to me there is a definite possibility that during some parts of the cycling operation moisture could have been re-evaporated from the cooling coil and condensed temporarily inside the subfloor ducts. Was this possibility investigated?

Then I was also interested in the heat lags that are quoted. As heat lag was defined in the paper, it was the difference in time between the occurrence of the maximum temperature on the outside and the maximum temperature on the inside of any given building component. There were also heat flow meters on these components which indicated the time at which they were transmitting maximum heat. I was primarily interested in the fact that the heat flow meter on the ceiling showed that there was practically no lag in this ceiling construction that had a full thick batt of rock wool in it. One would expect that a ceiling containing 3⅝-in. of rock wool would have several hours lag, but Table 4 shows that the lag was never more than half an hour, as indicated by the heat flow meter. Do the authors have any explanation for that?

In Table 3, Item 17, which is the net cooling load derived from the condenser heat rejection, is probably a better measure of the actual net cooling load than the values derived from the mass air flow and enthalpy change in the cooling coil because of the difficulty in accounting for moisture that was condensed.

This paper would be subject to a better analysis if it contained more data on outdoor temperatures at night, indoor air temperatures at night, register temperatures at night for continuous fan operation and inside wall and floor surface temperatures throughout the day.

I would like to ask whether *intermittent* or *continuous* blower operation resulted in less electrical consumption per degree day of cooling load. It is probable that cycling the blower and compressor together kept the earth below room temperature day and

night for the most part and undoubtedly caused the earth to be a smaller load in the daytime. Continuous circulation of air, taking advantage of the flywheel effect of the concrete slab and the earth during the late evening, might possibly cool the house below 75 F at night and would probably increase the load on the cooling unit when it first started in the morning.

A. B. NEWTON, Wichita, Kans.: Table 3, line 13, gives the temperature rise of water in the condenser. This value together with gallons per minute is used to calculate the heat rejection. I notice one peculiarity, and this may be associated with the way that the capacity was changed: the rise varies quite a bit. It goes from 12.7 deg for the early test to as high as 35 deg for the later tests.

One would expect that the discharge pressure would be some fixed amount above the temperature or the pressure corresponding to that leaving water temperature. It would seem that the net pressure was permitted to vary.

As a matter of information it might be well to include the suction pressures and the discharge pressures for some of these tests.

C. M. ASHLEY, Syracuse, N. Y.: What success have the authors had in attempting to establish a heat balance between the calculated heat balance from the heat flow meters and that from the equipment? I wanted to ask particularly about heat storage with the floor. Obviously, the floor acts as an excellent sink in a house of that sort. However, the ability to absorb depends upon whether it is covered with rugs or not.

With a single-story building such as this, most of the heat enters in the form of radiant energy; a form which the floor picks up very readily and for that reason acts particularly well as a sink. It appears that over a period of time the floor temperature has not risen—so that the calculation of the heat gain over a substantial number of days should indicate whether there is any net difference between the two different methods of calculation.

AUTHOR'S CLOSURE (H. T. Gilkey): Dr. Gilman inquired concerning the effect on the calculated cooling load had we assumed that the south wall was completely shaded. Obviously the calculated heat flow through the wall would have been decreased, and it is our feeling that the values would have been very close to those which were actually measured. Changing the method of calculation would involve a rather complicated adjustment of the sol-air temperatures for the south wall, and as a consequence, the heat gain was calculated assuming that the south wall was unshaded.

The velocity of the air leaving the register in the living room was approximately 350 fpm. Because of the perimeter-loop type of system in which the air flowed through a duct which was parallel to the long axis of the floor registers, it is to be expected that the register-air velocity will vary considerably from one end of the register to the other. On an average, however, these velocities were approximately 350 fpm. We do intend to use floor diffusers during future cooling investigations in the Research Residences.

Mr. Avery asked how the cooling unit capacity was reduced and why. All we did was to install the unit as it was delivered to us and we found that the capacity was down. Actually, we were not disappointed at this, even though we were a bit perplexed. We had a unit which was rated at 2 tons, the smallest central unit then on the market. We knew that it would be oversized, but had decided previously that we would just have to do the best we could with the oversized unit, and consequently we were not concerned about the lowered capacity. Of course, this lowered capacity threw some other things out of line, and we did not compensate for it. We had based our calculations upon the assumption that we could circulate 300 cfm of air per rated ton of refrigeration, or 600 cfm. In the case of the test series having the lower capacity, we wound up circulating 600 cfm per actual ton of refrigeration. This affected the temperature of the air leaving the coil, and I think explains why we

did not get as much dehumidification as might be expected from the long compressor operations. That is, the air was passing through the coil rapidly and the air temperature did not drop as much as it would had we been circulating the 300 cfm per ton.

This last comment may answer in part Mr. Achenbach's first question concerning why the relative humidity in the residence remained high despite the long compressor operations. He also asked about the discrepancy between the weighed amount of condensate and the amount which would be calculated from the humidity ratios shown in Table 3. In the first place, the wet bulb temperature in an air stream is rather difficult to measure, and we are not exactly sure how accurately we can measure it. Secondly, it takes time for moisture to build up in a coil, and unless a steady-state condition is achieved the water has a tendency to flow off the coil in fits and starts.

Mr. Achenbach also mentioned that one-fourth to one-third of the air temperature rise between the coil and the rooms occurred in the subfloor plenum, the feeders, and the perimeter loop. He has the feeling that the house is therefore not getting full benefit of the flywheel effect because of heat gains to the floor slab from the ground. Although this is undoubtedly true, we do not know the extent of this heat gain from the ground. We also have some panel cooling effect as Mr. Ashley has pointed out. The amount of panel cooling would also be quite difficult to evaluate.

We observed no evidence whatsoever of condensation occurring in either the feeder or the loop ducts. It is very possible that some condensation could have occurred since continuous observations were not made on all sections of the feeder and loop ducts. No evidence of this was noted either during or after the studies.

The apparent time lags of heat flow through the ceiling were very low. Although the reason for this cannot be explained exactly, reference to Table 4 of the paper may be helpful. In the first place, the exterior surface of the ceiling was considered to be the attic side of the insulation. The predicted time lags for the roof and ceiling combination were 3 hr, but the actual time lags were much less. In addition, the actual time lags were for the ceiling only, and did not include the roof. It may be noted from Table 4 that the heat flow through the ceiling reached a maximum value before the exterior surface temperature reached a maximum value for Series A-1, and that for Series A-2 the temperature and heat flow reached their maximum at approximately the same time. In the remaining two series the maximum heat flow occurred one-half hour after the occurrence of the maximum exterior surface temperature. This would indicate that factors other than heat flow from the attic were affecting the ceiling surface temperature. Consequently, it is our feeling that the term *apparent time lag* should be used since there are radiation and convection effects from other surfaces within the room.

Mr. Achenbach commented that if we deduct the pickup in the ducts from the total capacity of the unit, we find an above floor cooling load of 8,000 Btu per hr for the two series with continuous blower operation, and 12,000 Btu per hr when the blower was cycled with the compressor. Again, let me point out that this may be due in part to panel cooling effect. We do not know the effect of the type of blower operation on electrical cost per degree day of cooling. Actually, the term *degree day* has not yet been defined for cooling, and before we can go too far in accurate prediction of the cost of cooling residences such a definition must be made.

We have a different cooling situation in a residence than in any other type of construction. In the first place, the occupancy is much less than in other types of buildings. This means that the major part of the cooling load is due to the outdoor weather. In addition, in the residence we have more outside wall and roof area per sq ft of floor area than we do in other types of construction. Again, this would indicate that the weather will have a greater effect upon the cooling load in residential construction than it will with buildings designed for other uses.

Mr. Newton commented on the condensing cooling water flow rates and stated that the water temperature rise through the condenser varied from 12 to 35 F. The low temperature rise during the early studies was due in part to an overcharge of refrig-

erant on the system. This was one thing that occurred from our experimenting when we first got the cooling unit.

Mr. Ashley asked if we had been able to make any correlation of a heat balance, using heat flow meters and the heat balance around the cooling unit itself. We have not attempted to do this, since it has been pointed out very ably by several earlier authors that heat gain is not cooling load. Secondly, the heat flow meter measures only the heat flow through the walls and the ceiling. As we all know, windows and glass in general are very big factors in residential cooling loads. The floor of the residence was not covered; it was bare concrete.

1506

No. 1506

A TWO-SPHERE RADIOMETER

By D. J. SUTTON* AND P. E. MCNALL, JR.,** MINNEAPOLIS, MINN.

THE RESULTS of many investigations indicate that radiation accounts for almost half of the heat loss from a sedentary individual.[1] Since this is apparently true, radiation effects have received increasing attention in recent years, especially with the advent of panel heating and the large picture window type of construction.

The environmental mean radiant temperature, which is the main environmental factor influencing the human body's radiation heat loss, might be defined as that uniform surface temperature, of a black-body enclosure, which will result in the same net radiation heat loss, or gain, from the body under consideration as occurs in the actual environment being investigated. As yet, methods of measuring this factor are inconvenient to use. Lack of progress in developing better mean radiant temperature measuring devices is understandable when it is realized that the Mean Radiant Temperature (MRT) of a non-uniform environment will vary from point to point within that environment, and also will be different for objects of different shapes located at the same point. The difficulty of obtaining a sensing element capable of assuming the many shapes and positions of the human body is at once realized, and although several cylindrical radiation sensitive devices have been used to simulate the human body more accurately,[2,3,4,5] a spherical sensing element such as the globe thermometer is probably the only satisfactory instrument used at the present time to measure nondirectional MRT. The term nondirectional implies that the radiation influence on the element is equal in all directions.

The Globe Thermometer: Globe thermometers[6,7,8] are usually 6-in. or 8-in. diameter blackened spheres. Such a sphere is located at the point where the MRT measurement is desired, and the sphere equilibrium temperature is recorded along with the ambient air temperature and air velocity near the globe. These factors can then be substituted into an equation which, when solved, yields the nondirectional MRT. MRT determinations with the globe thermometer, therefore, require simultaneous measurement of three variables and solution of an equation. The equation has been made available in the form of an alignment chart, but nevertheless, the measurement procedure is not as simple as might be desired. An instrument capable of measuring nondirectional MRT inde-

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[1] Exponent numerals refer to Bibliography.

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pendently of air temperature and air velocity effects would be an improvement over the existing method. The instrument described in this paper was designed as an attempt at such an improvement.

A NEW NONDIRECTIONAL RADIOMETER*

The proposed two-sphere radiometer is simple and makes use of two geometrically identical spherical elements (Fig. 1). One sphere is gold plated and polished to minimize its radiation exchange by virtue of a low surface

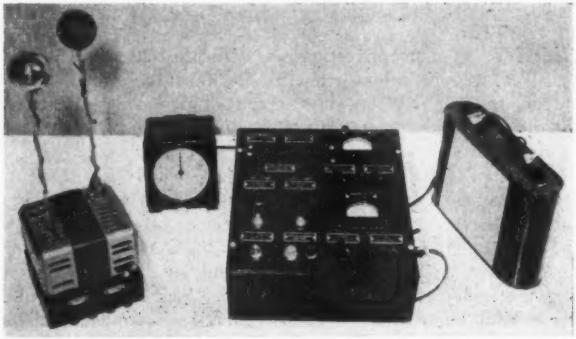


FIG. 1. TWO-SPHERE RADIOMETER AND ITS ASSOCIATED EQUIPMENT

emissivity, while the other has a blackened, high-emissivity surface to maximize its radiation exchange. The two spheres are electrically heated to some temperature which is usually above both the ambient air temperature and MRT, and are then controlled at that predetermined temperature. Since the spheres are

NOMENCLATURE

q = heat flow, Btu per hour.
σ = Stefan-Boltzmann constant, 0.173×10^{-8}, Btu per (hour) (square foot) (Fahrenheit degree absolute to the fourth power).
F_e = emissivity factor for radiation heat transfer, dimensionless.
F_A = angularity factor for radiation heat transfer, dimensionless.
A = area, square feet.
T = temperature, Fahrenheit absolute.

Subscripts

$_b$ = black sphere.
$_p$ = polished sphere.
$_i$ = input.
$_c$ = convection.
$_k$ = conduction.
$_r$ = radiation.
$_{aux}$ = auxiliary.
$_s$ = sphere.
$_{MRT}$ = mean radiant temperature.

* After a good share of the development work on this instrument was completed an article describing a radiometer based on a similar operating principle came to the authors' attention.[9]

geometrically identical, operate at the same temperature, and are located in essentially the same environment, the conduction and convection losses are the same for both, and the difference in heat inputs required to maintain the spheres at the same temperature is equal to the difference in radiation heat losses. Knowing this difference and the sphere operating temperature, the effective non-directional mean radiant temperature (MRT) of the surrounding environment can be obtained from a graph or by calculation. When the sphere temperatures are controlled, the difference in power inputs is a direct indication of

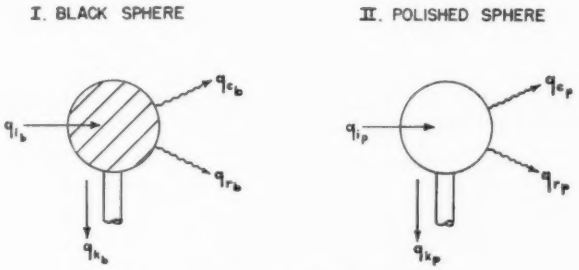


FIG. 2. SPHERE HEAT BALANCES

MRT. The control temperature and the physical constants (the sphere surface area and the sphere surface emissivities) must also be known.

Theoretical Analysis: When two geometrically identical spheres are heated to the same elevated temperature in identical environments, the heat balances indicated in Equations 1 and 2 will exist (Fig. 2).

$$q_{ib} = q_{cb} + q_{kb} + q_{rb} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$q_{ip} = q_{cp} + q_{kp} + q_{rp} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Since the spheres are geometrically identical, operate at the same temperature and are in the same environment, the conduction and convection losses must be the same for both.

$$q_{cb} + q_{kb} = q_{cp} + q_{kp} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Subtracting Equation 2 from Equation 1 and simplifying,

$$q_{ib} - q_{ip} = q_{rb} - q_{rp} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The Stefan-Boltzmann equation for computing net radiation heat exchange between two surfaces at temperatures T_1 and T_2 is[10]

$$q_r = \sigma F_e F_A A [T_1^4 - T_2^4] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

For the special case where a small sphere radiates heat to a large enclosure with an equivalent black-body surface temperature (T_{MRT}), the emissivity factor F_e can be assumed to equal the sphere surface emissivity, and the angularity factor F_A can be taken as unity.[10] When this is the case, the equations for the

sphere net radiation heat losses are as follows:

$$q_{rb} = \sigma F_{eb} F_{Ab} A_b [T_b^4 - T_{MRT}^4] \quad (6)$$

$$q_{rp} = \sigma F_{ep} F_{Ap} A_p [T_p^4 - T_{MRT}^4] \quad (7)$$

Since $F_{Ab} = F_{Ap} = 1$, $A_b = A_p = A_s$, and $T_b = T_p = T_s$, subtracting Equation 7 from Equation 6 and substituting into Equation 4 gives

$$q_{ib} - q_{ip} = \sigma A_s (F_{eb} - F_{ep}) [T_s^4 - T_{MRT}^4] \quad (8)$$

The quantity $(q_{ib} - q_{ip})$ is the difference in heat inputs to the two spheres and is hereafter referred to as auxiliary heat input with the designation q_{aux}. Therefore,

$$q_{aux} = \sigma A_s (F_{eb} - F_{ep}) [T_s^4 - T_{MRT}^4] \quad (9)$$

Fig. 3 is a plot of Equation 9 for a two-sphere radiometer utilizing 2-in. diameter spheres with surface emissivities of $F_{eb} = 0.94$ and $F_{ep} = 0.02$. Inspec-

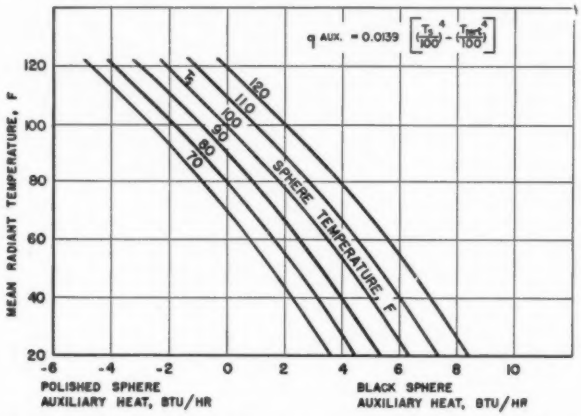


FIG. 3. MEAN RADIANT TEMPERATURE *vs.* AUXILIARY HEAT INPUT FOR VARIOUS SPHERE TEMPERATURES

tion of this figure indicates that when the sphere temperatures are controlled at some known constant value, the difference in power inputs (auxiliary heat) is a direct indication of the MRT.

DESCRIPTION OF THE EXISTING RADIOMETER

The first experimental radiometer model based on this operating principle is shown in Fig. 1, along with its control box, cycle timer, and auxiliary input battery. This model utilizes on-off temperature control of the sphere elements, and the average MRT is measured during a five minute interval. Two bellows type thermostats were modified so that the limit switch contacts provided small differential on-off temperature controls (Fig. 4).

The two 2-in. diameter spherical elements are actually thin copper shells (0.008-in. wall thickness). Each sphere contains two small manganin resistance

heaters and an iron-constantan thermocouple. These elements are mounted on four small hermetic seal units. The heaters are designated as primary and auxiliary heaters. The primary heaters are connected in parallel so that the primary inputs are the same to both spheres. The primary heat input is ordinarily controlled by the polished sphere thermostat so as to maintain that sphere at the desired control temperature. The primary input is usually not sufficient to bring the black sphere up to the control temperature, so some additional heat input is required. This additional input to the black sphere is controlled by the black sphere thermostat and is designated as auxiliary heat. The auxiliary heat brings the black sphere temperature up to the desired level and is a measure of the difference in radiation heat losses which can be related to MRT.

Using two heaters in this manner makes it possible to measure auxiliary input directly rather than by making two independent power input measurements and subtracting to obtain the difference. The auxiliary heater in the polished sphere would be used if the MRT were ever greater than the control temperature.

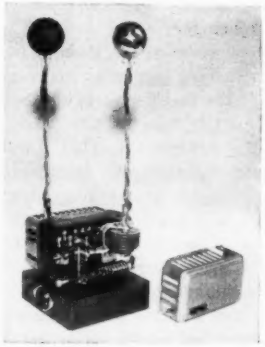


FIG. 4. RADIOMETER WITH THERMOSTAT COVER REMOVED

When this is the case, the primary heat input is controlled by the black sphere thermostat, and the polished sphere thermostat controls auxiliary heat input to the polished sphere. A primary heat selector switch on the control box makes it possible to reverse the control action. The iron-constantan thermocouples are located at the geometric centers of the spheres. These couples are used whenever it is desirable to adjust or check the control temperatures. Four binding posts on the base of the radiometer facilitate lead-wire connections to the temperature measuring instrument.

After final assembly the spheres and the thermostat control bellows were evacuated and charged with ethyl chloride. The quantity of fill is just sufficient to immerse the heating elements which extend approximately ¼ in. above the bottom of the sphere. The fill has three functions in the present radiometer model.

1. *The Control Function:* The liquid-vapor interface in the sphere senses sphere temperature, and the saturation pressure corresponding to this temperature is transmitted through the sphere mounting stem to the control bellows. The bellows movement actuates the thermostat contact linkage shown in Fig. 4.

2. *Prevention of Excessive Heater Temperatures:* The fill immerses the sphere heating elements and thereby prevents excessive heater temperatures

by providing a high heat transfer coefficient between the heater and the liquid.

3. Uniformity of Surface Temperature: The third function of the volatile fill is to provide uniformity of sphere surface temperature. The fill is continuously being evaporated from the liquid surface and condensing on the inside of the vapor exposed shell. This results in a uniform surface temperature with a relatively low heat capacity sphere. A low mass element is easier to control and has faster response to environmental changes than would be possible with a more massive element such as a solid copper sphere.

The spheres were mounted at a sufficient height above the radiometer base and at a sufficient distance from each other to reduce the effects of radiation interception to a negligible magnitude.

RADIOMETER CONTROL CIRCUIT

The radiometer control circuit and heater wiring diagram are shown in Fig. 5. A 6.3 volt filament transformer was used as the control circuit power supply and also supplies the power for the primary heaters. Two double-pole double-throw relays were used. A closure of the thermostat contacts in one direction completes the circuit to a relay and a green operation light. The energized relay in turn completes the circuit to the sphere heater, or heaters, and to a standard clock timer in the case of the auxiliary relay. A closure of the thermostat contacts in the reverse direction completes the circuit to a red operation light which gives an indication of temperature overshoot. The heat inputs should be adjusted to eliminate or minimize overshoot.

Two single-pole five-position selector switches provide a means of changing the resistance in series with the heaters and thereby varying the power input. Five steps of heater power were provided so that a wide range of environmental conditions could be covered.

Two polarized sockets were mounted in the side of the control box so that external connections could be made to battery power supplies. A 6 volt battery is used to supply auxiliary power, and as mentioned previously, the primary power is at present supplied from the control circuit transformer. A means of supplying battery power to the primary heaters was also provided. Two meters indicate d-c input voltages, and series potentiometers are provided so that the input voltages can be adjusted to the desired values. The radiometer was calibrated on the basis of an auxiliary input voltage of 5.0v d-c as indicated on the auxiliary input voltmeter.

The wiring diagram for the repeating cycler circuit is shown in Fig. 6. The circuit connections with the wiring of Fig. 5 are indicated. The cycler is housed in the control box and makes use of a synchronous motor-driven gear train and cam. The cam actuates a single-pole double-throw microswitch which acts in conjunction with a double-pole double-throw relay to provide a holding circuit. This holding circuit functions to produce a cycle of 5 min ± 1 sec duration every time the starter button is depressed. A white pilot bulb indicates that a cycle is in progress.

At the start of a cycle the microswitch cam follower is on the cam lobe and the switch is in position (a) as shown in Fig. 6. When the starter button is depressed the relay pulls in and remains energized through a holding circuit when the button is released. The holding circuit provides current to the cam

motor and pilot bulb through the *in* relay contacts. The cam motor rotates the cam, and after approximately 54 sec the microswitch follower leaves the cam lobe and the microswitch movable contact goes to position (b). When this occurs the relay is de-energized and the circuit to the cam motor and pilot bulb is completed through the *out* relay contacts. When the microswitch is in position (b) power is supplied to the standard timer every time the auxiliary input relay is energized. The microswitch stays in this position for 5 min ± 1 sec. The cam motor continues to drive, and when the cam reaches its original starting position the entire circuit is de-energized. The circuit cannot be reactivated until the starter button is again depressed. The Standard Timer

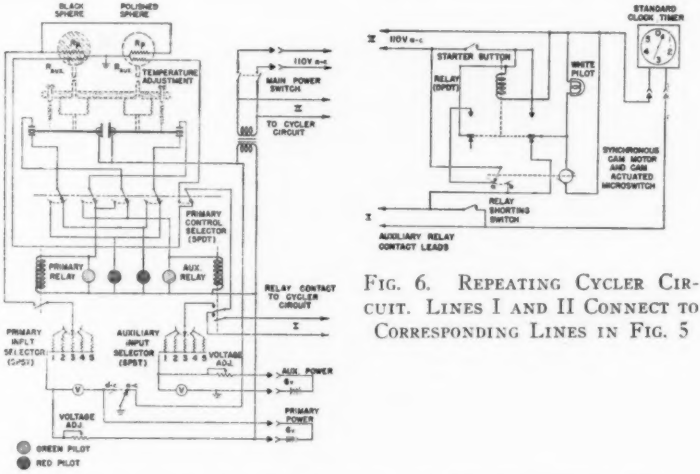


FIG. 6. REPEATING CYCLER CIRCUIT. LINES I AND II CONNECT TO CORRESPONDING LINES IN FIG. 5

FIG. 5. RADIOMETER CONTROL CIRCUIT AND HEATER WIRING DIAGRAM. LINES I AND II CONNECT TO CORRESPONDING LINES IN FIG. 6

(Fig. 1) plugs into an outlet on the front of the control box and runs every time the auxiliary relay is energized during the 5 min portion of the cycle. At the end of the 5 min interval the *Total-On-Time* reading is an indication of the average auxiliary power input during the 5 min period.

Fig. 7 shows the calibration curve for the radiometer operating with on-off control at a 90 F temperature. Mean Radiant Temperature is plotted against *Total-On-Time* for each of the five auxiliary input selector positions.

RADIOMETER OPERATION

To make an MRT measurement the operator adjusts the radiometer power inputs so that the sphere heaters cycle properly. Cycling is indicated by on-off operation of the green pilot bulbs on the control box. Having done this he depresses the cycler starter button. The on-cycle is indicated by a white pilot bulb. The operator is then free to perform other tasks during the measure-

ment interval. After approximately 5 min have elapsed, the cycler automatically turns off and the *Total-On-Time* reading is retained on the timer. When the white pilot goes out, indicating completion of the cycle, the operator records the *Total-On-Time* reading and the auxiliary input selector position. Knowing these terms the MRT can be obtained from the calibration chart, Fig. 7. The auxiliary input selector position will usually not have to be changed in any one environment. After recording the data for a particular run, the radiometer can be moved to a new position for another MRT reading. A 12-ft connection cable makes it possible to move the radiometer over a large area without having

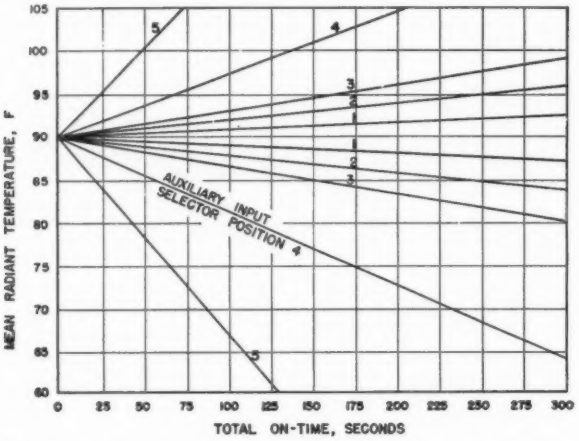


FIG. 7. RADIOMETER CALIBRATION CURVE FOR A 90 F CONTROL TEMPERATURE

to move the control box. To start the next measuring cycle the operator resets the timer to zero and depresses the starter button.

FIELD EXPERIENCE WITH THE TWO-SPHERE RADIOMETER

The radiometer has been used in several limited MRT surveys in homes and apartments in the Minneapolis area, and also was used to measure MRT in the A.S.H.V.E. Environment Laboratory in Cleveland, Ohio. Readings obtained in the Minneapolis area were reasonable in all cases and reproducibility of data was good. Readings obtained in the Environmental Laboratory agreed closely with MRT values computed on the basis of globe thermometer data, being within ± 0.4 F under two different sets of conditions. The MRT's in the Environment Laboratory were also computed on the basis of geometry and surface temperature measurements. The radiometer values and these computed values agreed within ± 1.5 F. Fig. 8 shows a typical radiometer test setup.

A critical error analysis indicated the importance of accurate temperature control. For this reason, a portable potentiometer has been used to make frequent checks of the sphere control temperatures. This minimizes control tem-

perature errors such as those caused by variations of atmospheric pressure. It is believed that the MRT measurements have been accurate to within ± 2 F.

It has been found in practice that the effects of visible and solar radiation are very important in some instances and should be considered. It is possible that these effects can be separated from the low temperature radiation effects, but the desirability of this separation is controversial from a comfort measurement viewpoint. The best method of handling these effects is yet to be determined.

FUTURE DEVELOPMENT

A considerable amount of work has been directed toward developing radiometers which will be smaller in physical size, have faster response to



FIG. 8. TYPICAL RADIOMETER TEST SETUP

environmental changes, and which will have an improved temperature control system. These advancements will be made possible by improved construction and heater design techniques, and by the availability of small composite hermetic seal units.

One model which is planned will feature electric heating of only a single sphere. In this case, the difference in equilibrium temperatures of the spheres will be measured and heat added to the cooler sphere to bring it up to the equilibrium temperature of the other. If the MRT is below air temperature, it will be necessary to add heat to the black sphere. If, on the other hand, the MRT is above air temperature it will be necessary to heat the polished sphere. This operating principle will eliminate primary heating and minimize the temperature difference between the spheres and the MRT. Minimizing this difference reduces the MRT error potential of the instrument. With the proposed instrument it will be necessary to measure both sphere temperature and heat

input to determine the MRT, but a simple method of combining these two factors into a single approximate indication is being considered. It is hoped that the instrument can be used in conjunction with a recording potentiometer to give a continuous MRT indication.

ACKNOWLEDGMENT

Acknowledgment is made of the advice and assistance provided by staff members of the Minneapolis-Honeywell Research Center where this project was carried on.

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DISCUSSION

A. J. HESS, Los Angeles, Calif.: This paper illustrates the tremendous amount of work and effort that people in the research end of our profession will unselfishly expend to develop a device that is only going to be used to get information and does not have many commercial applications. This type of work should be carefully reviewed and the authors congratulated for their efforts. They deserve our gratitude as this is often their only reward.

G. V. PARMELEE, Cleveland, Ohio: We at the Research Laboratory have been very much interested in the progress and development of this new radiometer. The problem of measuring mean radiant temperature is a specter that raises its head every time we work in the Environment Laboratory or become concerned with human comfort and radiation.

I think the authors have done an excellent job in applying the principle of polished and black spheres to a very simply operated instrument. They have made many improvements since we first saw it in Cleveland.

It might be of interest to this group to know something of the activity of other people in this same problem of mean radiant temperature measurement. The British Medical Research Council's Environmental Hygiene Research Unit, under Dr. Thomas Bedford, has applied this same principle to two metal men, as they call them. They are man-sized objects which are hollow, heated, provided with two internal fans to obtain uniform temperature distribution and have some twenty-two thermocouples for measuring surface temperature. The control panel for these "men" is a rather bulky thing. The entire arrangement is not nearly as portable as the instrument that Messrs. Sutton and McNall have developed, but it does give the mean radiant temperature with respect to a full sized man. I think the British will be reporting on this instrument in the near future.

JOHN EVERETTS, JR., Philadelphia, Penna.: This question of measuring mean radiant temperature is of vital importance to us in our evaluation of environmental conditions to reduce the so-called recognized degree of discomfort. The bibliography at the end of this paper gives a list of ten papers which have been presented on such things as radiometers, black sphere thermometers etc., which we have never been able to fully utilize in this work of comfort.

The instrument described in this paper may give us a tool through which we can get the information needed.

I would like to ask the authors several questions in connection with this particular instrument.

(1) What is the effect of a point source of high temperature radiation, such as an incandescent light on the evaluation of the black and the polished sphere?

(2) In the paper the solar radiation effect has not been taken into account. How important is this if one is taking mean radiant temperatures in the presence of a south window, for example?

(3) Of what significance is the geometry of the room with respect to the size of the sphere?

(4) I believe there was some discussion several years ago on the size of the sphere with relation to the size of the room in which the mean radiant temperatures were taken. With this instrument is the reading in a large room as accurate as that in a smaller room?

(5) We know that mean radiant temperature not only has an effect on the human body from the standpoint of temperature, but also from the standpoint of moisture. Some years ago in the laboratory Dr. F. C. Houghten did some very preliminary work on the use of a eupatheoscope or a radiometer, which was enclosed in a wetted wick to determine the effect of mean radiant temperature on moisture. Has this been explored with this type of radiometer, and if so, what has been the effect?

L. F. SCHUTRUM, Cleveland, Ohio: This instrument is a special contribution to the field of panel heating and panel cooling; however, I do have one thought concerning the estimated error. I suspect that measurements of mean radiant temperatures made with this radiometer are more precise than the ± 2 F indicated by the authors.

AUTHOR'S CLOSURE (P. E. McNALL): We are interested in better comfort control and we feel that an instrument of this type which is relatively simple to use will produce data on the elusive radiation effect which will enable the industry to produce better comfort controls, better heating and cooling system components and evaluate the various newer types of building construction. The instrument was developed as a possible aid to this problem. Its details have been presented at the meeting so that others interested in this problem may also use our experiences to further their work along this line. The entire task is much too far reaching for any one person,

laboratory or company, and we believe that only concerted effort by many can make the headway necessary for better understanding of the radiation problem.

AUTHOR'S CLOSURE (D. J. Sutton): We appreciate the thanks for our work expressed by the discussers. It is gratifying to hear that there are many people who feel that an instrument such as this is a worthwhile contribution to the heating and ventilating industry.

It is interesting to hear Mr. Parmelee's comments on some of the investigations that he ran across in England.

Mr. Everetts asked a question as to the effect of a point source, such as an incandescent light, on radiation-measuring devices. A high temperature point source will have a considerable effect if it is placed in the immediate vicinity of the instrument, *i.e.*, within 6 to 12 in. As soon as the point source is placed any appreciable distance away from the non-directional radiometer, the solid angle factor for radiation heat transfer between the sphere and the point source under consideration will be so small that in most cases this effect will be negligible.

As for the effect of solar radiation, that is a question that is unanswered. It depends upon many factors including the amount of solar radiation coming in through a window and the color of the clothing that one is wearing. You can see that it is difficult to attempt to compensate for unpredictable factors such as these.

It is possible that the low temperature radiation effects could be separated from the high temperature solar effects, but the desirability of making this separation is somewhat dubious from a comfort standpoint. Solar radiation enters through a window and if it strikes the body it has an immediate effect. If it doesn't strike the body, it is absorbed by some other surface and it later occurs in that environment as a low temperature radiation effect. Thus it is there twice. It is just a matter of how it is to be handled, and as yet, we don't know how we want to handle it.

As for the relative size of the sphere and the room, I am sure that in any normal sized room the sphere is so much smaller than the enclosure that the configuration factor can always be assumed to equal unity.

I believe that I have read of instruments using a wetted wick when attempting to measure a combined effect. They were not trying to isolate the radiation effect, but rather measure the combined effect of all environmental factors on an object. By using the wetted wicks they attempted to compensate for the relative humidity effect.

Mr. Schutrum commented that our instrument was probably much more accurate than we claim. We cannot determine the exact accuracy because we have no method of checking the calibration. The instrument is calibrated on the basis of well established theory, but our accuracy is no better than the assumptions which were made. We are working on a calibration chamber which will be completed soon and at that time it may be possible to give you accurate figures which are better than those which we have stated in the paper.

Our literature survey indicated that there were more instruments available for measuring directional effects than for measuring the non-directional effects. Therefore, we have directed our efforts toward development of an instrument which could be used for measuring the non-directional mean radiant temperature. The same principle could be applied for directional instruments.

1507

No. 1507

SEMI-ANNUAL MEETING, 1954

SWAMPSCOTT, MASSACHUSETTS

THE Semi-Annual Meeting, 1954, held at the New Ocean House in Swampscott, Mass., attracted a total of 568 members, their wives and guests. Arrangements for the meeting were very capably handled by the Committee on Arrangements composed of Massachusetts Chapter members. Registration totals for the three-day period, June 28-30, 1954, were as follows: 287 members, 186 ladies, 56 guests.

The first session was called to order by President L. N. Hunter in the Ballroom of the New Ocean House, at 9:30 a.m., Monday, June 28. General Chairman R. T. Kern welcomed the members on behalf of the Massachusetts Chapter.

A Committee on Resolutions was appointed by President Hunter consisting of B. L. Evans, St. Louis, Mo.; J. H. Ross, Toronto, Ont., Can.; and I. W. Cotton, Indianapolis, Ind., Chairman.

President Hunter called upon First Vice Pres. J. E. Haines to present a proposed amendment to the By-Laws. This section now reads as follows:

ARTICLE VII—Committees

Section 3(h). Committee on Research (paragraph 5)

The Committee on Research shall, as soon as may be practicable after the close of each Annual Meeting of the Society, recommend to the Council for appointment a *Director of Research*, whose activities, proceedings and reports shall be subject to the direction of the Committee and the approval of the Council. The salary of the Director of Research shall be fixed by the Council upon the recommendation of the Committee on Research, and his employment may be terminated in the Council's discretion.

To be amended as follows:

The Committee on Research shall recommend to the Council for appointment a *Director of Research* whose activities, proceedings and reports shall be subject to the direction of the Committee and the approval of the Council. The salary of the Director of Research shall be fixed by the Council upon the recommendation of the Committee on Research, and his employment may be terminated in the Council's discretion.

It was moved by Mr. Haines, seconded and voted unanimously that Article VII, Section 3(h), para. 5 be amended, the amendment to be presented to the membership for action at the 61st Annual Meeting in accordance with Article IX of the By-Laws.

Pres. Hunter then introduced Nathaniel Glickman to act as chairman of the technical discussions. Three papers were presented by their authors (see program, p. 311).

At the Second Technical Session on Tuesday, June 29, 1955, at 9:30 a.m., First Vice Pres. John E. Haines called the meeting to order and turned the session over to Prof. G. L. Tuve.

Prof. Tuve introduced the authors of three of the technical papers, each of whom presented his paper. A fourth paper was presented by title only. Following the discussions the meeting was adjourned by First Vice President Haines at 12:30 p.m.

John W. James, second vice president, called the Third Technical Session to order and introduced C. S. Leopold as chairman. Mr. Leopold called upon the authors to present their papers. There were three papers presented and discussions followed each of the presentations.

At the request of Second Vice President James, I. W. Cotton presented the report of the Committee on Resolutions as follows:

RESOLVED:

That the attending members of the Society, their families and guests, voice appreciation and acknowledge deep gratitude:

1. To Massachusetts Chapter for the genuine warmth of its hospitality;

2. To the Committee on Arrangements for its outstanding success in providing for our every comfort and pleasure;

Arranging so thoughtfully and meticulously, in advance, for air conditioning the entire area for our activities instead of limiting the salubrities to the rooms in which we have deliberated and gamboled;

Offering us humidities ranging from negligible to 100 percent to make it easy for our technical experts to expound our theories with full examples right there at hand;

Offering us an excursion which included an engine failure at sea for the edification of the uninitiated and permitted them to demonstrate the monumental calm which all such disasters should engender, and finally and most importantly in

Seeing to it that we always had skilled bus drivers on our excursions who navigated us through horrendous traffic hazards and brought us back to home base with miraculously whole skins and without benefit of life preservers.

3. To the New Ocean House Management and Staff for catering so delightfully to our culinary and refreshment urges and for furnishing spotless linens for our infrequent and regrettable lapses into repose;

4. To our Hosts for inviting such excellent speakers for the Welcome Luncheon and the Semi-Annual Banquet;

5. To the Officers of the Society, the technical staff, the Committee members, the research workers and speakers who cooperated in presenting interesting and significant papers and also to the members who spiced the Technical Sessions with questions and comments;

6. To the multitude of conscientious Committee Members who so intrepidly embraced stern duty and labored ever and anon in the Society and public interest in the face of beckoning and almost irresistible distractions;

7. To the welcoming host who proved to us over the loud speaker that the artistry of state or sectional ballyhoo can no longer be considered to thrive exclusively on its native Texas range; and last but by no means least

8. To everybody participating in this valuable and gala meeting for adding to the serious accomplishment and the grand good times by injecting their vivid personalities and inspiring friendliness into a get-together based on the zenith of human relationships—A.S.H.V.E. service and camaraderie.

Respectfully submitted,
RESOLUTIONS COMMITTEE
I. W. COTTON, *Chairman*
BRUCE W. EVANS
JOHN H. ROSS

On motion of Mr. Cotton, seconded by J. M. vanNieukerken, the resolutions were adopted unanimously.

The session was adjourned at 12:00 noon by Second Vice President James.

The Fourth Technical Session got underway at 2:00 p.m. Wednesday, June 20th, with Pres. L. N. Hunter presiding. The authors of the papers scheduled for presentation at this session were introduced by President Hunter. Each in turn presented his paper.

There was no business brought up at the session's conclusion and the meeting was adjourned by President Hunter at 4:30 p.m.

PROGRAM SEMI-ANNUAL MEETING

New Ocean House, Swampscott, Mass.—June 28-30, 1954

Friday—June 25

10:00 a.m. Finance Committee *(Suite D—Main Floor)*

1:30 p.m. Executive Committee *(Suite D—Main Floor)*

Saturday—June 26

10:00 a.m. Council Meeting *(Standish Room—Lower Foyer)*

1:30 p.m. REGISTRATION *(Lobby Floor)*

2:30 p.m. Research Executive Committee, R. S. Dill, *Chairman (Suite D—Main Floor)*

Sunday—June 27

9:00 a.m. Bus and Boat Trip to Provincetown, Mass.

1:30 p.m. REGISTRATION *(Lobby Floor)*

1:30 p.m. Committee on Research, R. S. Dill, *Chairman (Priscilla Room—Main Floor)*

2:00 p.m. Musicale and tea

2:00 p.m. Motor trips to nearby historical points.

Monday—June 28

9:00 a.m. REGISTRATION *(Lobby Floor)*

9:30 a.m. FIRST TECHNICAL SESSION *(Ballroom)*

Call to order by Pres. L. N. Hunter
Nathaniel Glickman, *Chairman*
Greetings by R. T. Kern, *General Chairman*
Response by President Hunter
Amendments to By-Laws

Physiological Response to Sudden Changes in Atmospheric Environment: Studies of Normal Subjects, Obese, Hyperthyroid and Hypothyroid Patients, by Tohru Inouye, F. K. Hick, M.D., R. W. Keeton, M.D., and Lionel Bernstein, M.D., Chicago, Ill., presented by Dr. Hick

A Rapid Heat Meter Thermal Conductivity Apparatus, by J. D. Verschoor and Ada Wilber, Manville, N. J., presented by Mr. Verschoor

Monday—June 28 *(Continued)*

FIRST TECHNICAL SESSION *(Continued)*

Irradiation of Vertical and Horizontal Surfaces by Diffuse Solar Radiation from Cloudless Skies, by G. V. Parmelee, Cleveland, Ohio, presented by the author

12:15 p.m. WELCOME LUNCHEON *(Dining Room)*
Toastmaster: A. L. Hare
Speaker: Edward Rowe Snow, *Author*
Subject: New England Legends

1:30 p.m. Committee on Research, R. S. Dill, *Chairman (Priscilla Room—Main Floor)*

1:30 p.m. GOLF TOURNAMENT *(Tedesco Country Club)*
Research, Eichberg and Paul Bunyan Cups—18 Hole Medal Play

1:30 p.m. Deep Sea Fishing Trip—Swimming at beach

2:00 p.m. Guide Committee, R. E. Cherne, *Chairman (Presidents' Room—Main Floor)*

7:30 p.m. TAC Coordinating Committee, John Everetts, Jr., *Chairman (Presidents' Room—Main Floor)*

7:30 p.m. TAC on Odors, T. H. Urdahl, *Chairman (Gold Room—Main Floor)*

7:30 p.m. TAC on Sensations of Comfort, C. S. Leopold, *Chairman (Standish Room—Lower Foyer)*

7:30 p.m. BOSTON "POPS" CONCERT (Symphony Hall, Boston) Arthur Fiedler conducting. Engineer's Night at the "Pops"

Tuesday—June 29

9:00 a.m. REGISTRATION *(Lobby Floor)*

9:30 a.m. SECOND TECHNICAL SESSION *(Ballroom)*
Call to Order by First Vice Pres. John E. Haines
G. L. Tuve, *Chairman*

Characteristics of Downward Jets of Heated Air from a Vertical Discharge Unit Heater, by Linn Helander, S. M. Yen and L. B. Knee, Manhattan, Kans., presented by Dr. Yen

Computing Temperatures and Velocities in Vertical Jets of Hot or Cold Air, by Alfred Koestel, Cleveland, Ohio, presented by the author

Sound-Pressure Levels and Frequencies Produced by Flow of Water Through Pipe and Fittings, by W. L. Rogers, Evanston, Ill., presented by the author

Computation Charts and Theory for Rectangular and Circular Jets, by H. G. Elrod, Jr., Cleveland, Ohio, presented by title

10:00 a.m. LADIES PARTY—*Oriental Rugs and the Stories They Tell*, by Arthur Gregorian

1:30 p.m. Long-Range Planning Committee *(Presidents' Room—Main Floor)*

1:30 p.m. Nominating Committee Meeting *(Standish Room—Lower Foyer)*

1:30 p.m. TAC on Air Distribution, W. O. Huebner, *Chairman (Gold Room—Main Floor)*

1:30 p.m. TAC on Heat Flow Through Glass, R. A. Miller, *Chairman (Priscilla Room—Main Floor)*

1:30 p.m. TAC on Weather Data, John Everetts, Jr., *Chairman (Jade Room—Main Floor)*

Tuesday—June 29 *(Continued)*

1:30 p.m. Boat and Bus Trip to Salem and Gloucester

1:30 p.m. Historical Tour to Boston

3:30 p.m. TAC on Hot Water and Steam Heating, John W. James, *Chairman (Suite E—Main Floor)*

4:00 p.m. Chapters Conference Committee, W. J. Collins, Jr., *Chairman (Colonial Room—Lower Foyer)*

7:00 p.m. NEW ENGLAND SHORE DINNER *(Dining Room)*

7:30 p.m. TAC on Heating Loads, P. R. Achenbach, *Chairman*

7:30 p.m. TAC on Panel Heating and Cooling, P. B. Gordon, *Chairman (Presidents' Room—Main Floor)*

7:30 p.m. TAC on Sorption, G. L. Simpson, *Chairman (Gold Room—Main Floor)*

8:30 p.m. Woodland Fears—Real or Imaginary—Gilbert E. Merrill, Ass't. Director of Education, Boston Museum of Science
Informal Dancing

Wednesday—June 30

9:00 a.m. REGISTRATION *(Lobby Floor)*

9:30 a.m. THIRD TECHNICAL SESSION *(Ballroom)*
Call to Order by Second Vice Pres. John W. James
C. S. Leopold, *Chairman*
Filtration of Microorganisms from Air by Glass Fiber Media, by H. M. Decker, J. B. Harstad, F. J. Piper and M. E. Wilson, Frederick, Md., presented by Mr. Decker
Effects of Room Size and Non-Uniformity of Panel Temperature on Panel Performance, by L. F. Schutrum and J. D. Vouris, Cleveland, Ohio, presented by Mr. Schutrum
Cooling Load from Sunlit Glass and Wall, by C. O. Mackey and N. R. Gay, Ithaca, N. Y., presented by Prof. Mackey
Report of Committee on Resolutions

10:00 a.m. Ladies Visit to noted North Shore Gardens

11:30 a.m. Ways and Means Committee, J. C. Fitts, *Chairman (Presidents' Room—Main Floor)*

2:00 p.m. FOURTH TECHNICAL SESSION *(Ballroom)*
Call to Order by Pres. L. N. Hunter
Cooling Studies in a Research Home, by W. S. Harris and P. J. Waibler, Urbana, Ill., presented by Prof. Harris
Analogue Computer Analysis of Residential Cooling Loads, by T. N. Willcox, C. T. Oergel, S. G. Reque, C. M. toeLaer and W. R. Brisken, Bloomfield, N. J., presented by Mr. Willcox
The Operating Cost of Residential Cooling Equipment, by S. F. Gilman, L. A. Hall, and E. P. Palmatier, Syracuse, N. Y., presented by Dr. Gilman
Adjournment

7:00 p.m. SEMI-ANNUAL BANQUET *(Dining Room)*
Toastmaster: Past Pres. Lauren E. Seeley
Presentation of Past President's Memory Book to Reg. F. Taylor
Speaker: Rev. Lex King Souter, Minister, First Congregational Church, Fall River, Mass.
Subject: Your Future is Now

COMMITTEE ON ARRANGEMENTS

R. T. KERN, *General Chairman*
A. L. HARE, *Vice Chairman*
L. E. SEELEY, *Honorary Chairman*

Banquet: G. D. Fife, *Chairman;* E. L. Blair, R. E. Reid, F. C. Williams.

Entertainment: L. R. Geissenhainer, *Chairman;* M. I. Kishler, W. H. Krapohl.

Finance: W. H. Shipp, *Chairman;* A. I. Brayman, R. F. Curry, A. M. Lovenberg, G. E. MacNeill, G. E. Roberts, Jr., D. J. Stressenger, H. L. Von Rehberg.

Ladies: D. M. Archer, *Chairman;* Mrs. D. M. Archer, Mr. & Mrs. John Bonner, Mrs. J. N. Gates, Mrs. A. L. Hare, Mr. & Mrs. L. V. Richards, Dean & Mrs. L. E. Seeley, Mr. & Mrs. R. G. Vanderweil.

Publicity: J. MacG. Means, *Chairman;* A. I. Brayman.

Sports: W. G. Burbo, *Chairman;* R. C. Berry, W. G. Martin, Jr., D. W. Noble, J. N. Russo, F. W. Thorley.

Reception: G. W. Sprague, *Chairman;* J. W. Brinton, J. H. Colby, A. A. Fava, G. E. Fickett, H. L. Flanagan, M. I. Kishler, C. W. Larson, F. A. Merrill, H. E. Parker, E. M. Tracey, W. A. Williams.

Sessions: John Bonner, *Chairman;* J. N. Gates, J. P. Hoar, J. A. Ivester, A. L. Nelson, J. M. Pettingell, B. H. Snow, R. B. Stevens, R. J. Tesar, R. G. Vanderweil.

Transportation: H. L. Farrow, *Chairman;* W. F. Austin, I. S. Dane.

1508

No. 1508

PHYSIOLOGICAL RESPONSES TO SUDDEN CHANGES IN ATMOSPHERIC ENVIRONMENT

Studies of Normal Subjects, Obese, Hyperthyroid and Hypothyroid Patients

By TOHRU INOUYE,* F. K. HICK,** M.D., R. W. KEETON,† M.D. AND LIONEL BERNSTEIN,†† M.D., CHICAGO, ILL.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and by the United States Public Health Service in cooperation with the University of Illinois, College of Medicine, Chicago, Ill.

THIS REPORT discusses the physiological adjustments made upon sudden entrance and departure from a hot, humid environment by women who have probable abnormalities in the regulation of body temperature. The data may be compared with those of young, healthy women.[1] This study was undertaken to determine alterations in physiological responses in the presence of handicaps on heat loss or heat production. Alterations in the responses would provide an understanding of means by which the body adjusts heat loss under adverse environmental conditions. Interference of heat loss is suggested in the obese patients. Their thick layer of subcutaneous fat has been regarded as insulation somewhat comparable to an overcoat. On the other hand, abnormalities in heat production are observed in hyperthyroid and hypothyroid patients. The high heat production of the hyperthyroid would require a high heat loss, and the low heat production of the hypothyroid a low heat loss, to maintain normal body temperature.

METHOD

Young, healthy women, nurses from the Research and Educational Hospitals, were used as controls for this study. Subjects with abnormalities in their heat regulation were selected from the medical services of the Hospital. The physical measurements for all subjects are summarized in Table 1. The oxygen consumption measured by a clinical instrument at the beginning of a three hour test averaged 121 cc/(min) (m^2) for the healthy, 131 cc/(min) (m^2) for the obese, 181 cc/(min) (m^2) for the hyperthyroid and 99 cc/(min) (m^2) for

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[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1954.

TABLE 1—PHYSICAL CHARACTERISTICS AND AGES OF INDIVIDUAL SUBJECTS

SUBJECT CLASSIFICATION	EXPERIMENTS	AGE	HEIGHT, IN.	WEIGHT LB	SURFACE AREA SQ FT
NORMALS					
E.F.	[a]	24	67	129	18.2
P.K.	[a]	24	60	115	15.9
M.L.	[a]	22	68	122	17.8
D.M.	[a]	23	71	169	21.1
J.M.	[a]	23	64	120	17.0
T.M.	[a]	22	64	117	16.8
M.O.	[a]	26	67	120	17.5
C.C.	[b]	19	63	124	16.9
J.D.	[b]	24	64	112	16.4
C.H.	[b]	21	63	107	16.0
G.K.	[b]	24	66	116	16.9
M.L.	[b]	19	64	135	17.9
L.M.	[b]	20	66	150	19.2
D.My.	[b]	21	66	146	18.4
D.N.	[b]	20	62	105	15.5
HYPERTHYROID					
E.S.	[a] and [b]	35	63	154	18.6
M.B.	[a] and [b]	36	62	104	15.6
E.J.	[a] and [b]	27	70.5	138	19.6
E.K.	[a] and [b]	43	68	133	18.3
S.G.	[a] and [b]	24	64	138	18.0
E.W.	[b]	18	65.5	118	17.1
Z.H.	[b]	23	66.5	133	18.2
J.C.	[b]	54	63.5	185	20.2
L.D.	[b]	42	66	114	16.9
G.G.	[b]	61	60.5	125	16.6
S.H.	[b]	44	64	145	18.3
HYPOTHYROID					
M.F.	[a] and [b]	43	61	116	16.1
M.P.	[a]	39	66.5	146	18.9
E.L.	[a]	42	67	166	20.1
L.L.[c]	[a] and [b]	60	67.5	157	19.7
OBESE					
E.F.	[a] and [b]	38	65	174	20.1
M.P.	[a] and [b]	37	65	234	22.9
C.P.	[a] and [b]	24	64	213	21.7
E.S.	[a]	22	66.5	181	20.7
S.P.	[a] and [b]	43	62.0	354	26.2
M.G.	[a] and [b]	21	64.5	172	20.1
C.H.	[a]	29	67.0	194	21.5
A.H.	[a]	50	64.0	280	24.3
C.V.	[b]	35	64.0	330	26.3
C.M.	[b]	45	64.0	247	23.2

[a] Subjects participated in experiments in which they were moved from a comfortable to a hot humid room and subsequently returned to the comfortable.
[b] Subjects participated in experiment in which they remained in a comfortable environment (76 F, 30 percent RH) for three hours.
[c] Male.

the hypothyroid patients. Assuming all oxygen consumption with a respiratory quotient of 0.82 is converted to heat, these rates of oxygen consumption would correspond to 13.3 Btu/(hr) (sq ft), 14.0 Btu/(hr) (sq ft), 19.2 Btu/(hr) (sq ft), 10.6 Btu/(hr) (sq ft), for the healthy, obese, hyperthyroid and hypothyroid patients, respectively. All subjects reported to the building after a light breakfast or lunch. They rested about ½ hr before entering the test room.

The subjects in this study were dressed in long union suits reaching to the ankles and to the wrists. The union suit was of 90 percent cotton and 10 per-

TABLE 2—SUMMARY OF AVERAGES OF MEAN SKIN TEMPERATURE, RECTAL TEMPERATURE, THERMAL SENSATION AND HEAT LOSS BY VAPORIZATION FOR NORMAL, HYPERTHYROID AND OBESE SUBJECTS KEPT FOR THREE HOURS IN A ROOM AT 76 F AND 30 PERCENT RH

TIME OF OBSERVATIONS	SUBJECT CLASSIFICATION		
	Normal	Hyperthyroid	Obese
	Mean Skin Temperature—F		
60 min[a]	91.2	92.4	91.5
120 min	90.3	92.0	91.3
180 min	90.0	91.7	90.8
	Rectal Temperature—F		
60 min	98.6	98.8	98.8
120 min	98.6	98.5	98.5
180 min	98.6	98.5	98.5
	Thermal Sensations[b]		
60 min	4.1	4.2	4.0
120 min	4.0	4.1	4.0
180 min	4.1	4.1	4.0
	Heat Loss by Evaporation—3rd Hour		
Btu/(hr) (sq ft)	4.35	7.58[c]	3.77

[a] Time from entrance into test room.
[b] Thermal sensations are numerically described in which 4 = thermal neutrality (comfortable), 5 = slightly warm, 6 = warm and 7 = hot; 3 = slightly cool, 2 = cool and 1 = cold.
[c] Indicates significant differences of this group average from the normal.

cent wool, knitted fabric. The suits were rigged with thermocouples to permit observations of skin temperature at 19 locations and the rectal temperature. In addition to these measurements, body weights were observed accurately to 0.02 oz, and thermal sensations were recorded.[2]

The study was divided into two types of tests to determine the manner of heat loss under approximate *steady states* in a comfortable environment and to determine any deviations of these partitions upon sudden change in the temperature and relative humidity of the environment. When the patients were studied for heat loss under approximate *steady states*, they remained seated for three hours in a comfortable environment of 76 F ±1 deg F dry bulb with a 30 ±2 percent relative humidity (RH). When these patients were studied for responses to sudden change in temperature and humidity, they remained for one hour in a comfortable room (76 F dry bulb with a 30 percent RH) immedi-

ately followed by an exposure for one hour in a hot humid room (98.5 F dry bulb with a 66 percent RH) and then returned to the comfortable room for another hour.

RESULTS

Fig. 1 and Table 2 present observations on normal, obese and hyperthyroid women kept three hours in a comfortable environment. The hyperthyroid

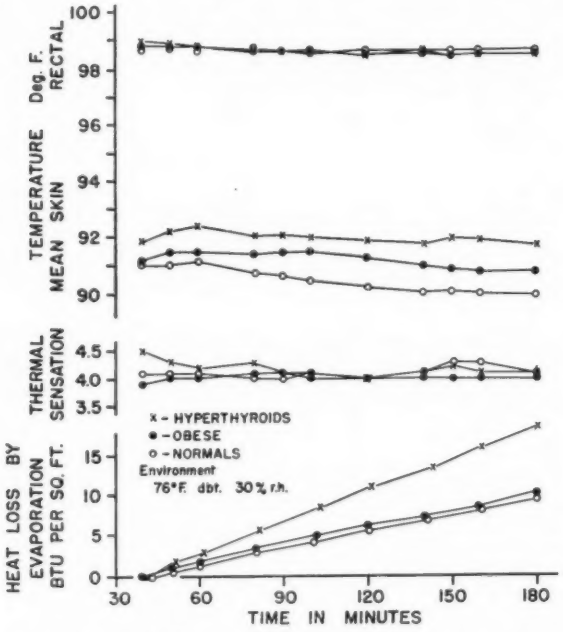


FIG. 1. THE MEAN SKIN TEMPERATURE, THE RECTAL TEMPERATURE, THERMAL SENSATION, AND HEAT LOSS BY EVAPORATION IN HYPERTHYROID, OBESE, AND NORMAL WOMEN DURING A THREE-HOUR EXPOSURE AT 76 F, 30 PERCENT RH

subjects showed persistently higher mean skin temperatures than the normal women. Their heat loss by evaporation was also greater. The rectal temperatures were maintained in each of the three groups with no significant difference between the groups. The hyperthyroid and obese women reported during the three hours of exposure thermal sensation of *comfortable* similar to the young healthy women. Sufficient data on hypothyroid patients are not available in this regard.

Fig. 2 and Table 3 compare the adjustments of normal, obese, hyperthyroid and hypothyroid women after entry into a hot, humid room from a comfortable

room. The warming of the skin in all women was prompt. The hyperthyroid women showed the most rapid rise in skin temperature. This warming reached a plateau within 10 min reflecting early onset of free sweating. All subjects raised their skin temperatures about 5 deg in 10 min after entering the hot room. Immediately upon entry to the hot, humid room, all subjects felt warm. They attained skin temperatures within 0.5 deg of 97 F in about 30 min of heat

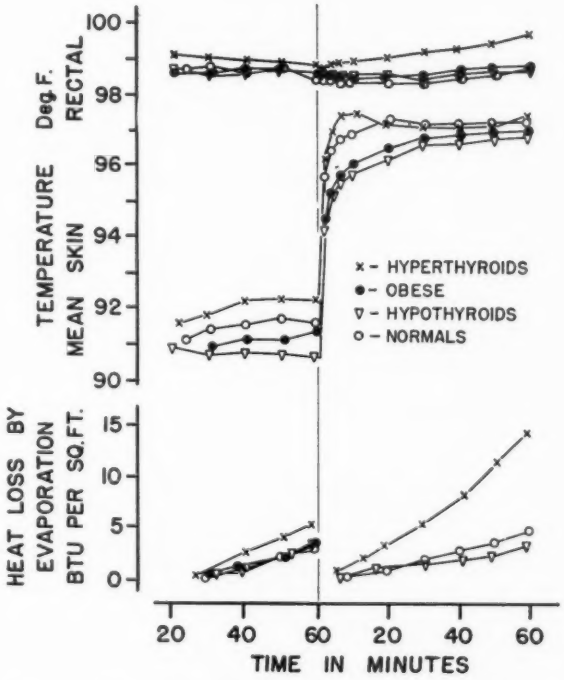


FIG. 2. THE MEAN SKIN TEMPERATURE, THE RECTAL TEMPERATURE, AND THE HEAT LOSS BY EVAPORATION IN HYPERTHYROID, OBESE, NORMAL, AND HYPOTHYROID WOMEN ON ENTERING A HOT HUMID ROOM (98.5 F, 66 PERCENT RH) FROM A COMFORTABLE ROOM (76 F, 30 PERCENT RH)

exposure. Only the hyperthyroid women showed steady rises in rectal temperature significantly above that of the other subjects. Five other hyperthyroid patients were not able to endure the heat exposure for more than 30 min.

Table 3 shows the average time for sweating for each group. The normal women required the longest time to sweat; whereas, the obese and hyperthyroids began to sweat promptly on entrance to the hot humid room. Table 4 presents the data on sweating for each subject in more detail. The time of onset of sweating varied widely for individuals. It is apparent that the hyperthyroid women sweat the most, and the hypothyroid sweat the least. The profuse

TABLE 3—GROUP AVERAGES OF MEAN SKIN (MT_s) AND RECTAL TEMPERATURE (T_r), HEAT LOSS BY EVAPORATION AND THERMAL SENSATION (TS) FOR EACH GROUP OF SUBJECTS, FIRST IN THE COMFORTABLE ROOM AND THEN IN THE HOT HUMID ROOM

OBSERVATIONS	SUBJECT CLASSIFICATION			
	Normal (7)[a]	Hyper-thyroid (5)	Hypo-thyroid (4)	Obese (8)
COMFORTABLE ROOM				
76 F, 30 percent RH				
After 60 min				
MT_s (F)	91.6	92.2	90.6	91.2
T_r (F)	98.4	98.8	98.5	98.5
TS	4.0	4.4	4.0	3.9
Heat Loss by Evaporation Btu/(hr) (sq ft)	4.4	8.5[b]	5.0	5.6
HOT HUMID ROOM				
98.5 F, 66 percent RH				
On Entrance				
TS	6.1	6.4	6.2	6.7
After 10 min				
MT_s (F)	96.8	97.4	95.6[b]	96.0
T_r (F)	98.3	98.9	98.5	98.4
Change in MT_s in 10 min F deg	+5.2	+5.2	+5.0	+4.8
Change in T_r in 10 min F deg	−0.1	+0.1	0.0	−0.1
Time of Onset of Sweating (minutes)	15.6	6.5	13.4	4.1[b,d]
Amount of Sweat in Hour				
In ounces	4.8	8.4[b]	2.3[c]	6.3
In oz/sq ft	0.27	0.49[b]	0.12[c]	0.22

[a] Number of subjects in a group.
[b] Indicates significant difference of this group average from the normal.
[c] Indicates strongly suggestive difference of this group average from the normal.
[d] One patient required 59 min to sweat. This value placed her statistically outside of the group; hence, her value was excluded from the average.

sweating of the hyperthyroids is reflected in their high rate of evaporation. shown in Fig. 2.

Fig. 3 and Table 5 summarize the adjustments of the normal, obese, hyperthyroid and hypothyroid women upon entry into the comfortable room after heat exposure. The rectal temperatures of all subjects rose slightly, followed by a decline to similar levels by the end of one hour. The skin temperatures fell promptly. The skin temperatures were similar for the normal and hyperthyroid women; they were the same for the hypothyroid and obese women. Cooling by evaporation of sweat in the clothing was greatest for the hyperthyroid and least for the hypothyroid women. The obese women felt cooler than the normal

women upon leaving the hot humid room. There appeared to be no difference in the thermal sensations by the end of the hour; they felt comfortable or very close to comfortable.

The evaporation of sweat has been held to explain the adjustments of heat loss mechanisms during recovery from heat exposures. In comparing the similarities of skin temperatures of the normal and hyperthyroid women it becomes apparent that the greater evaporation in the latter subjects is accompanied by a rapid lowering of elevated rectal temperature to normal levels, *i.e.*, 99.7 F to 98.8 F. On the other hand, the slow cooling of the skin of the hypothyroid women is immediately related to their small amount of evaporation.

TABLE 4—TABULAR PRESENTATION OF SWEAT DATA ON INDIVIDUAL SUBJECTS

SUBJECTS	COMFORTABLE ROOM BEFORE HOT ROOM	HOT ROOM		COMFORTABLE ROOM AFTER HOT ROOM	
	Btu/(hr) (sq ft)	Time of Onset of Sweating, min	Sweat oz/(hr) (sq ft)	Accumulated Sweat oz/sq ft[a]	Heat Loss by Evaporation Btu/(hr) (sq ft)
NORMALS					
E.F.	3.7	17	0.32	0.25	11.9
P.K.	5.0	10	0.20	0.20	15.1
M.L.	4.5	4	0.15	0.14	10.6
D.M.	5.1	35	0.27	0.24	14.2
J.M.	3.0	10	0.39	0.33	13.3
T.M.	5.3	31	0.10	0.11	9.7
M.O.	4.5	3	0.45	0.34	9.6
HYPER-THYROID					
E.S.	11.0	4	0.21	0.11	15.2
M.B.	13.9	3	0.61	0.25	22.5
E.J.	7.0	5	0.36	0.07	11.4
E.K.	4.3	14	0.57	0.29	18.4
S.G.	6.2	8	0.69	0.35	24.4
HYPO-THYROID					
M.F.	5.5	5	0.10	0.07	9.5
M.F.	5.6	3	0.22	0.10	9.9
E.L.	4.7	32	0.14	0.07	5.1
L.L.[b]	4.2	14	0.02	0.04	6.0
OBESE					
E.F.	6.2	10	0.33	0.19	17.1
M.P.	3.9	2	0.26	0.13	10.8
C.P.	3.6	2	0.15	0.06	9.6
E.S.	4.1	5	0.06	0.19	14.2
S.P.	7.8	2	0.003	0.02	6.4
M.G.	4.5	59[c]	0.22	0.01	11.7
C.H.	6.3	7	0.46	0.23	15.1
A.H.	8.8	2	0.30	0.16	15.3

[a] At time of re-entry into comfortable room.

[b] Male.

[c] The analysis of data of obese women excludes this subject since 59 min for the time of onset of sweating is statistically not representative of this group of subjects.

The slow cooling of the skin of the obese women can be explained perhaps by the warmed subcutaneous fat supplying heat to the skin. This is justified by the similarity of rectal temperatures and evaporation for the normal and obese women having comparable heat productions.

CONCLUSIONS

1. In comfortable environments under approximate *steady state*, high heat production (hyperthyroids) is balanced by a high rate of heat loss by evaporation and a slightly higher skin temperature than normal women. Obesity does not interfere with heat loss in environments which are comfortable for young, healthy women.

2. The physiological adjustments of hyperthyroid, hypothyroid, obese and healthy young women to sudden warming during exposure to a hot humid environment are

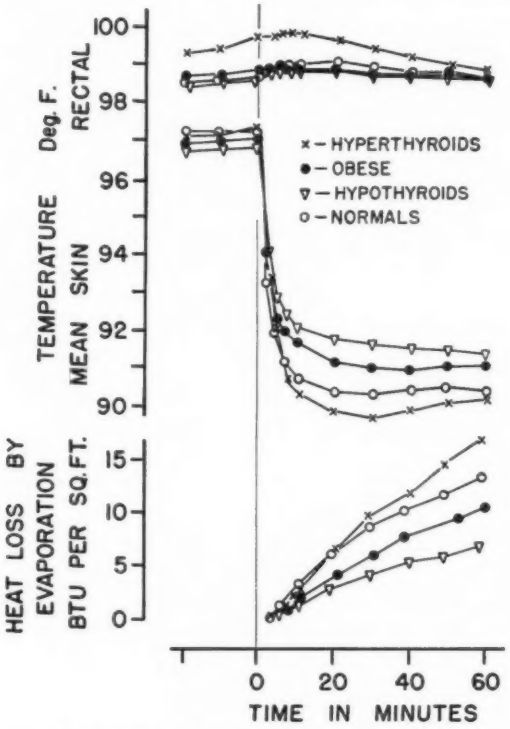


FIG. 3. THE MEAN SKIN TEMPERATURE, THE RECTAL TEMPERATURE, AND THE HEAT LOSS BY EVAPORATION IN HYPERTHYROID, OBESE, NORMAL, AND HYPOTHYROID WOMEN ON ENTERING A COMFORTABLE ROOM (76 F, 30 PERCENT RH) FROM A HOT HUMID ROOM (98.5 F, 66 PERCENT RH)

TABLE 5—GROUP AVERAGES OF MEAN SKIN (MT_s), RECTAL TEMPERATURE (T_r), HEAT LOSS BY EVAPORATION AND THERMAL SENSATION (TS) FOR EACH GROUP OF SUBJECTS, FIRST IN THE HOT HUMID ROOM AND THEN AFTER RE-ENTRY INTO THE COMFORTABLE ROOM

OBSERVATIONS	SUBJECT CLASSIFICATION			
	Normal (7)*	Hyper-thyroid (5)	Hypo-thyroid (4)	Obese (8)
HOT HUMID ROOM				
98.5 F, 66 percent RH				
After 60 min				
MT_s (F)	97.2	97.3	96.8	96.9
T_r (F)	98.7	99.7[a]	98.7	98.8
TS	6.7	6.9	6.9	6.9
COMFORTABLE ROOM				
76 F, 30 percent RH				
At Entrance				
Accumulated Sweat (oz/sq ft)	0.23	0.22	0.07[a]	0.12[a]
TS	2.1	2.6	1.5	1.4[a]
After 10 min				
MT_s (F)	90.8	90.6	92.1	91.8
T_r (F)	99.0	99.8[a]	98.8	99.0
Change in MT_s in 10 min	−6.4	−6.7	−4.7[a]	−5.1[a]
Change in T_r in 10 min	+0.3	+0.1	+0.1	+0.2
After 60 min				
MT_s (F)	90.2	90.2	91.3	91.1
T_r (F)	98.6	98.8	98.6	98.7
TS	3.9	4.1	3.2	4.1
Heat Loss by Evaporation In one Hour				
Btu/(hr) (sq ft)	13.6	18.4[b]	7.6[a]	12.5

[a] Indicates significant difference of this group average from the normal.
[b] Indicates strongly suggestive difference of this group average from the normal.
* Number of subjects in group.

made promptly despite handicaps in heat regulation. Obese and hyperthyroid women sweat readily. All hyperthyroids, however, are not able to endure long exposures to heat. Their heat production exceeds their heat loss evidenced by a significant rise in rectal temperatures.

3. The physiological adjustments of hyperthyroid, hypothyroid, obese and healthy young women to sudden cooling after heat exposure depends greatly upon the evaporation of sweat collected in the clothing and upon the temperature of the body (heat content). The cooling of the skin is exaggerated by increasing the rate of cooling by evaporation, *i.e.*, the quantity of sweat trapped in the clothing. Cooling by evaporation is increased by high body temperatures. The presence of large amounts of subcutaneous fat slows skin cooling during recovery from heat exposure.

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2. A.S.H.V.E. RESEARCH REPORT No. 1447—Physiological Adjustments of Clothed Human Beings to Sudden Change in Environments: First Hot Moist and Later Comfortable Conditions, by F. K. Hick, M.D., Tohru Inouye, R. W. Keeton, M.D., Nathaniel Glickman and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 189).

DISCUSSION

W. A. GRANT, Fayetteville, N. Y.: This is another one of the papers from the University of Illinois which is a real contribution to the Society.

In providing for human health and comfort the air conditioning industry is interested in those people who deviate from the normal. This paper indicates that people whose reactions might deviate substantially are able to adjust rather readily.

Concerning the hyperthyroid can the authors tell how much change in rectal temperature under hot room conditions is considered sufficiently important so that real effort is warranted in making these people more comfortable in the hot weather?

Another question is what proportion of the total population is hyperthyroid to such a degree that air-conditioning in the hot summer weather will really be useful to them.

Are there many of the type of people who were unable to cooperate simply because they were unhappy, these again being in this hyperthyroid group?

H. E. ZIEL, Detroit, Mich.: This interesting paper recalled the problem of high temperatures to which some workmen are subjected in industry. Along with the discomfort experienced, a workman may readily reach a point of fatigue which may lead to bodily injury.

At the presidential safety meeting in Washington on May 4, 1954, there was considerable discussion on the maximum temperatures that should be allowed in a plant. We know the difficulties in cooling a plant or in even making it reasonably comfortable especially where equipment on processes radiate heat and the men must be close to the work.

I would like to know if some information could be formulated that would help the architects and engineers in the design of a building and its mechanical equipment which would enable them to reduce temperatures in the hot areas.

In the last fifteen years considerable changes have been made in types of buildings. They are made larger with side walls a great distance from the center of the building; the roofs are solid, blackout type and do not have the monitors and sawtooth construction as formerly.

I think it would be of interest if information could be compiled which would give a time or a temperature limit to which a workman should be exposed.

C. S. LEOPOLD, Philadelphia, Pa.: This excellent paper and the two papers of reference provide excellent source material and demonstrate that entering or leaving an air conditioned space is not harmful.

There has been in the past, and still exists to some degree today, the thought that there is a relation of indoor and outdoor temperature in summer. The only basis for this assumption was that it sounded plausible. Many practitioners in this art have considered that the concept was fallacious and these papers go a long way in justifying their conclusion.

The idea has been quite tenacious, possibly because an inadequate system would compensate for the outdoor condition and it was, therefore, possible to make a virtue of necessity.

R. W. MCKINLEY, Pittsburgh, Pa.: I believe that the authors should devote some effort to a study of physiological reactions of children to changes in environment.

The need for better design data for use in the large and active school building program is apparent. There is little information available, and it is not widespread

or well understood. I would like the authors to consider including the school age population in their plans for further study.

AUSTIN WHILLIER, Cambridge, Mass.: When the authors have the patients sitting on the scale are they reading or engaged in any active thought, such as studying a text book or are they asked to sit, close their eyes and relax?

In my limited experience, intense mental activity seems to increase the metabolic rate.

W. M. WALLACE, II, Durham, N. C.: We have had quite a number of discussions with members of the medical profession in regard to the temperature to be maintained in operating rooms. I would like to ask Dr. Hick if he could comment on or give any recommended temperature that should be used with the required 55 percent relative humidity. This question has been argued by engineers and doctors and there is no authentic answer at the moment.

R. A. MILLER, Pittsburgh, Pa.: I wonder whether there is any means for measuring the moisture (sweat) absorbed in the clothing. Could it be measured by electrical conductance of the clothing or some other means?

NATHANIEL GLICKMAN, Miami Beach, Fla.: There are certain points of interest which I think should be mentioned. Dr. Hick pointed one out, that under comfortable conditions the hyperthyroid patients had a greater evaporative heat loss than the normal subjects. Newberg and others have shown the direct relationship of insensible perspiration and basal metabolism; although these subjects were not basal, the hyperthyroid patients had a metabolic rate about 44 percent higher than the normal subjects in this particular series of experiments.

The authors state that for the 3-hr exposures in a comfortable environment the hyperthyroid subjects showed a persistently higher mean skin temperatures than the normal women. This appears to be so, but Table 2 does not show that the differences are statistically significant.

Furthermore, the short exposure experiments in Fig. 2, the differences between the two groups, that is, the normal and the hyperthyroid, do not appear to be significant.

From the value shown in Table 3 I am inclined to question the significance of the statement that *the hyperthyroid women showed the most rapid rise in skin temperature.*

It is more likely that the normal subjects had a more rapid rise than the obese subjects during the first ten minutes in the hot room. Dr. Hick mentioned that in his presentation. The fact that the rectal temperature of the hyperthyroid subjects rises more than does the rectal temperature of the other subjects is of marked importance.

These subjects were storing heat. In fact, they had developed a fever at the end of the hour in the hot room and were heading for trouble. Their temperatures had gone above 99.6 F, at about which point we had earlier found that the oxygen consumption or the heat production of individuals started to go up, as did their cardiac output.

Dr. Hick emphasized this in reporting that five other patients could not endure that heat for more than 30 min and it is proper to point out, as has already been done, that the hyperthyroid patients are noted for their lack of tolerance to heat.

The data also suggests to me that there might be a significant difference in the onset of visible sweating between the obese and normal subjects. It is possible that the author intended to use a superscript b instead of an a for the obese group in Table 3, which would indicate that the difference is significant.

The use of two different explanations for the slower cooling of the skin of the hypothyroid and obese subject when compared to the normal subjects is not necessary. I am inclined to disagree with the authors' explanation on obese subjects. Both the hypothyroid and obese groups had significantly less moisture available for evapora-

tion than the normal subjects on entrance into the comfortable room after the hot room exposure, as shown in the table. This has been shown before to be a most important factor affecting the response to the mean skin temperature under these conditions. Added to this is the lower heat production of the hypothyroid group.

Finally, the hyperthyroid patients presented a different problem. This group and the normal subjects have the same amount of moisture available for evaporation on entrance into the comfortable room, as Dr. Hick pointed out. Their mean skin temperature responded similarly in the comfortable room. Why, then, did the hyperthyroid subjects lose heat by evaporation more rapidly? They had developed a fever in the hot room which probably caused an increase in their already high heat production as well as cardiac output. They returned to the comfortable room with this augmented heat production.

Aside from the vapor pressure gradient, the higher production of heat facilitated the process of evaporation. As pointed out by Dr. Hick, this greater evaporation was accompanied by lowering of the rectal temperature. As the body cooled below the fever level the augmented heat production returned toward its earlier values.

We may ask what the results of this paper mean. It would appear that except for the effect of heat on the hyperthyroid patients the physiological adjustment of these patients to sudden changes in the physiological environment occurred rapidly and with relative ease.

R. C. EDWARDS, Pompton Plains, N. J.: Fig. 3 shows that when hyperthyroids went from the hot room to the cold room their mean skin temperature dropped to the lowest point. If the tests had been continued for another two hours, would the line indicating their mean skin temperature have crossed the others and returned to a point above them?

AUTHORS' CLOSURE (Dr. F. K. Hick): I might answer Mr. Edwards first. We have not carried out such an experiment but it seems likely that the mean skin temperature of the hyperthyroids would have risen above the others. It would take considerable time to do it and for various reasons experimental observations of this sort running over four hours are difficult.

Mr. Grant asked what proportion of the population is hyperthyroid. It must be very small. It would have no importance outside of hospital populations, I assume, or occasional family situations where it would be transient, presumably, corrected by treatment.

He also asks the question of how much change in rectal temperature warrants making these people comfortable? That is philosophic and one person may object strenuously to a temperature that another may endure without complaint.

A good many of these patients, as a matter of fact, do not object to ordinary situations in their daily living. Although we look upon them generally as people who are hyper-irritable and easily annoyed, if they live in Chicago in hot weather they are very tolerant.

Mr. Ziel mentioned hot spots in industry and the possibility that there should be some limiting time of exposure. We have done no work in this field and prefer to make no comment about it.

The question was asked about the need for standards for design data pertaining to children. It is well known that children produce heat at a greater rate than do adults per unit area, they are also extremely active and this activity produces additional heat. We have not planned any experimental observations for children. The data would be valuable, but test procedures would first have to be devised.

Usually the subjects of our experiments just sit without any apparent activity. They are not ordinarily permitted to read.

We have not yet observed any of them engaged in mental activity which induced sweating.

Mr. Wallace asked what temperature was recommended for an operating room if the relative humidity has to be fixed at 55 percent. I don't know and it would seem that one of the considerations in planning the temperature of the operating room is to satisfy the surgeon; the other is to make sure that the patient, who is commonly covered by drapes and exposed to the glare of an intense light, and commonly has five or six people standing at his side, will experience as little difficulty as possible in losing his body heat regardless of the dry-bulb temperature in the air-conditioned operating room. If data is to be collected on optimum operating room temperatures and humidities a simple device might be to maintain a continuous record of rectal temperatures for a few months in an operating room by thermocouples and a recording potentiometer which could be done relatively easily.

Mr. Miller asked about the method of measuring sweat in the clothing. Our method is to measure the patient's weight loss. I do know now that it can be done electrically. Some of you working in the clothing field may have other techniques for measurement.

One strange thing which can be done with clothing is to take a summer suit and move it without a man in it from a normal room into a 98 F, 66 RH room where it will pick up and ounce or two of weight by absorption. It does that very rapidly, I might add.

We appreciate very much the general spirit and friendliness of the discussions. However, at this time it seems to me that we have reached a point where we are not liable to get further major returns from the current type of studies. If we can devise some other means of measuring responses to heat, then there might be a very real opportunity for an entirely different line of approach.

We have held the opinion for a long time that for subjects who have to sweat freely, this must constitute some sort of physiologic stress or burden which they would be better off without, but we do not know how to prove that with some measureable data. As long as a subject can keep his rectal temperature normal that environment is physiologic for him as he goes into hotter stages, as near as I can tell. I am not talking about the cooler zones. We need to learn what degree of heat first constitutes stress, without showing a rise in rectal temperature, then we will have a tool for going further in some different manner. What that might be I can only surmise at this time.

Mr. Inouye is working on some chemical measurement of adrenal hormones and the like which shows sufficient promise to justify devoting a fair share of our finances to it for some years. We do not yet have any substantial data to indicate whether these research studies will be successful.

AUTHORS' CLOSURE (Tohru Inouye): The authors are gratified by the number of critical and provocative discussions of this paper.

The title of the paper indicates a presentation of results of a number of tests on patients for their reaction to several environments. The data from these patients have far reaching implications toward the progress of the study of heat regulation, particularly, under stress in a hot humid environment. It is apparent that the hyperthyroid and hypothyroid patients can make sudden changes in their heat regulatory mechanisms even with manifested handicaps. Their tolerances to any environment are predicated by their infirmity. However, given a selection of environment of their liking, these patients are capable of comfortably eliminating their body heat. This is also true of the obese patient.

This and preceeding works on physiological adjustments point the direction of study of man's reaction to the physical environment. I think we are agreed that most individuals can make sudden changes in environmental temperature and humidity as well as adjusting to a variety of combinations of temperature and humidity, physiologically speaking. Further, I am inclined to think that these changes are

actuated and maintained with changes in expended energy, thermodynamically speaking. The rate and control of the utilization of energy appear to be the integration of the peripheral, visceral, and central components of the heat regulatory mechanism involved in stress induced by sweating with or without fever as the result of exposure to a hot humid environment with or without work. Several direct means are currently in the developmental stage which should be invaluable parameters. It appears to be a logical and systematic approach to the problem of acclimatization to environment, fatigue and ill-effects of hot environments, asymmetric psychrometry (temperature, relative humidity and air velocity), and the role of age, sex, disease, and trauma as stress on heat regulation.

1509

No. 1509

A RAPID HEAT METER THERMAL CONDUCTIVITY APPARATUS

By J. D. VERSCHOOR* AND ADA WILBER,** MANVILLE, N. J.

THE NEED of an apparatus to determine the thermal conductivity of materials rapidly has become increasingly more apparent. Although guarded hot plate thermal conductivity apparatus (*A.S.T.M.* Test Method C-177)[1] leave little to be desired in the way of absolute accuracy of test results, they are extremely slow in operation and require the attention of skilled personnel. Frequently as long as two weeks are required to run a series of points on a single material with a guarded hot plate apparatus. As a result, the number of materials that can be tested in a period of time is very limited. This can be especially restricting for insulation development programs where statistically designed experiments require tests on as many as 16, 25, or even more different samples in order to obtain a reliable picture of the effect of a variable on conductivity.

The guarded hot plate apparatus can be speeded up somewhat by the incorporation of automatic balance controls.[2] However, the attainment of steady-state heat flow with the guard at precisely the same temperature as the test section is inherently a slow process.

Recently considerable work has been done with transient heat flow apparatus in which a slender probe is utilized.[3-6] With these apparatus data need be taken for only a very short period of time. A chart by Hooper and Chang[5] greatly simplifies the calculation of conductivity from these data. However, the construction of a probe is difficult. The ratio of the heated length of the probe to its diameter should be at least 100 to 1. Since it is desirable to keep the length short, of necessity the diameter becomes very small. D'Eustachio[3] mentions probes with an outside diameter of 0.030 in. and smaller containing a bifilar spiral heater winding and a thermocouple junction. Furthermore, because the rise in temperature of the probe during a test is small, accurate measurement of this temperature rise is difficult.

It was with this background that the authors undertook the development of a rapid conductivity tester. Speed, simplicity and precision without necessarily high absolute accuracy were the requirements. Of these simplicity was considered most important.

DESIGN

The upper half of each unit of the apparatus is raised by a crank so that test samples may be inserted easily and quickly. The temperatures within the

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[1] Exponent numerals refer to References.

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apparatus are all controlled automatically and do not require any attention from the operator. The only physical measurements needed are eight thermocouple potential readings and a thickness determination by means of a dial gage incorporated in the apparatus. By means of a single slide rule calculation involving a constant for the apparatus the thermal conductivity can be determined.

In order to achieve simplicity some overall accuracy was sacrificed. An example of this was in the assumption that the temperature difference across the sample varies directly with the difference in millivoltage between the hot- and cold-surface couples. As the response of a thermocouple is not exactly linear



FIG. 1. HEAT METER RAPID CONDUCTIVITY APPARATUS

with temperature, there is a slight error. The assumption made, however, permits a single potentiometer reading for temperature difference which eliminates two conversions with a thermocouple chart and a subtraction. Similarly the allowance of 1½ hr for steady-state heat flow may not be sufficient in all cases. However, if a testing procedure is strictly adhered to these inaccuracies will not affect the testing precision, and the conductivity of one sample may be accurately compared with another.

DESCRIPTION OF APPARATUS

The rapid conductivity apparatus (Fig. 1) consists of four units each capable of testing four samples in a working day. This amounts to 80 tests a week or a potential of over 4,000 a year. Basically each unit of the heat meter conductivity apparatus consists of a constant-temperature hot plate and a constant-temperature cold plate to maintain a temperature gradient through the test sample, and

a heat flow meter in series thermally with the test sample to measure the rate of heat transfer.

The size of test samples for the rapid conductivity tester was set at a nominal thickness of 1 in., 9 × 9 in. square minimum, 12 × 12 in. square maximum. The minimum size allows for an area surrounding the heat meter of about two times the thickness to minimize edge heat losses. These size limits also permit the testing of circular samples of 10 to 12 in. diameter. With the present apparatus, samples of 1 in. nominal thickness may vary between 0.9 and 1.3 in. Samples much thicker than 1.3 in. cause no mechanical difficulties in testing; however, the results are not always in good agreement with tests on regular samples. Under special conditions samples as small as ½ × 6 × 6 in. can be tested, although again these results are not always directly comparable with samples of standard size. For these thin sample tests a dummy aluminum plate ½ in. thick is inserted in one of the units.

Two of the test units, designated V and W, are designed to operate at a mean temperature of approximately 150 F; the other two units U and T at 350 F and 500 F mean respectively. The low-temperature units have steam-heated hot plates and cold plates cooled by tap water. Both of these provide constant temperatures without the use of expensive controls. Of course there is a seasonal variation of tap water of about 40 deg; however, the variation during any day and from one day to the next is small enough to be of little consequence. During the coldest part of the winter the mean temperature of the low-temperature units is in reality about 130 F instead of the nominal 150 F. The high-temperature units (350 F and 500 F mean) have steam-heated cold plates. The hot plates in these units are heated electrically by strip heaters and are maintained at constant temperatures of about 480 F and 780 F respectively by controllers.

In order to simplify plumbing connections each unit has the steam plate as the lower or stationary plate (Fig. 2). By mounting a heat flow meter directly on each steam plate in all the units, all the meters may be calibrated at substantially the same temperature, regardless of the temperature range of the unit in which they are used.

It is important that the steam be held at constant pressure to insure constant temperature. Any changes in the steam temperature affect the instantaneous rate of heat flow and lead to unstable readings. A precision pressure reducing valve with individual steam traps on each plate was not able to hold the steam temperature constant because of the action of the traps. Without the traps the plates operate at atmospheric pressure and so maintain a constant temperature. A mixer with a water-spray nozzle condenses excess steam from the plates without adding a pressure drop.

The upper plates in the units (hot surface for the high-temperature units, cold surface for the low) have grooves milled in the contact surfaces with thermocouples in these grooves to measure surface temperatures. Because it was impossible to cut grooves in the heat meters mounted on the lower plates, the temperatures of these surfaces are determined by fine wire thermocouples stretched across the heat meter. The hot- and cold-surface thermocouples are connected in series differential to simplify the operation of taking data.

The heat flow meters were of a standard type and consist of 360 pairs of silver-constantan thermocouples located between the two surfaces of a synthetic

resin wafer 3/64 × 4-1/2 × 4-1/2 in. A flow of heat through the wafer causes a small difference in temperature between the faces. The large number of thermocouples multiplies the minute voltage of a single pair of thermocouples across the small temperature difference to the point where it can be conveniently measured. While the heat meters are provided with a calibration curve by the manufacturer, it has been the practice to calibrate them periodically in a guarded hot plate. This is done at three temperatures near 212 F, and the value at 212 F interpolated. One heat meter was found to have a different calibration with heat flow in one direction than in the reverse direction. Accordingly it is necessary to calibrate and use the meters with the same heat flow direction. Experience over the past two years has shown that, with one exception, the

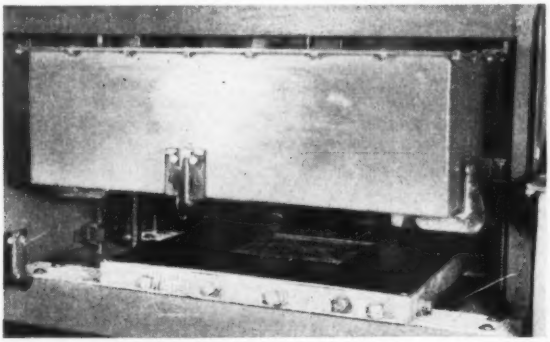


FIG. 2. CONDUCTIVITY APPARATUS OPEN SHOWING HEAT FLOW METER MOUNTED ON STEAM PLATE

calibrations have remained fairly constant with time. The calibration of the defective meter increased by a factor of 200 percent over a six month period. It was subsequently replaced by the manufacturer.

Each upper plate has an adjustable stud fastened to the top, as shown in Fig. 1. The distance between the top of the stud and the frame of the apparatus is directly related to the thickness of the sample in the apparatus. This means that a dial gage placed on the frame with its foot on the stud indicates the sample thickness when the length of the stud is adjusted properly.

A series of tests was made which indicated that with a test sample in place a heating time of 1½ hr was long enough to approximate steady-state heat flow conditions. Fig. 3 shows test sample in place. Additional time did not improve the test accuracy appreciably. Following the 1½ hr conditioning, four readings each of the heat meter potential and the temperature difference potential are taken at 5 min intervals. The quotient of the averages of these readings times the sample thickness and the calibration constant is the conductivity of the material.

The temperatures of 150 F mean, 350 F mean, and 500 F mean are representative ranges for many commercial insulations. To cover the higher-temperature insulations, a rapid conductivity apparatus of the heat meter type is currently being constructed at this laboratory. The range of this apparatus will include 1000 F mean temperature.

APPARATUS PRECISION AND ACCURACY

To determine the precision and accuracy of the heat meter apparatus, a test program was carried out to show (*a*) comparison between the results of a guarded hot plate apparatus (*A.S.T.M.* Test Method C-177)[1] and the heat meter on identical samples, and (*b*) the reproducibility of results possible with this apparatus.

For these tests three samples of different materials which had previously been tested in the guarded hot plate apparatus were chosen. These samples (Mineral wool insulation, designated A; Moderate-temperature insulating block, designated B; High-density insulating panel, designated C) were to represent typical materials that will be tested in the heat meter units.

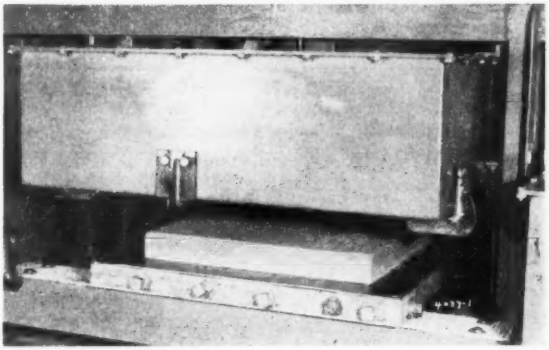


FIG. 3. CONDUCTIVITY APPARATUS OPEN SHOWING TEST SAMPLE IN PLACE

These samples were tested in rotation in each of the four heat meter units for four successive days. The results of this series of tests, given in Btu in. per (hour) (square foot) (Fahrenheit degree) are included in Table 1.

Comparison of these results with the conductivities as determined in the guarded hot plate apparatus showed that the heat meter values were consistently lower. The ratios of results, hot plate to heat meter, expressed in percentages ranged between 105 and 116, and averaged 111. The degree of error does not seem to be attributable to specific samples or to the heat meter unit used.

Instead of trying to compensate for these differences, it was decided to recognize that the apparatus is only fair in absolute accuracy and to concentrate on the reproducibility of results. Since materials submitted for determination of thermal conductivity by the heat meter apparatus are parts of tests designed to be evaluated by comparison, the properties of reproducibility and relative accuracy are much more important than absolute accuracy. To satisfy these conditions it was desired that there be both good duplication of results between units run at the same mean temperature (between units T and U and between V and W) and, more important, close repetition of results when a sample is retested in the same unit.

By statistical methods it was shown that no significant difference exists between the conductivities determined by the two 150 F mean heat meter units,

V and W. The same analysis showed that a small but significant difference does exist between the results of unit T run at 350 F mean and unit U for all samples tested. However, measures to counterbalance this were not attempted since unit T was designed primarily for use at 500 F mean.

Further analyses were performed to determine the reproducibility of results when the same sample is retested in the same heat meter unit. Standard deviations of less than 0.006 conductivity units were found for samples A and B. (The standard deviation s defines the range from the arithmetic mean $\bar{x}$, $(\bar{x} \pm s)$ within which approximately two thirds of the results would lie if an infinite

TABLE 1—REPRODUCIBILITY OF CONDUCTIVITY RESULTS

	Day	Thermal Conductivity Btu/(hr) (sq ft) (F deg/in.) Sample		
		A	B	C
Heat Meter V (150 F Mean)	1	0.260	0.344	0.601
	2	0.256	0.336	0.623
	3	0.259	0.334	0.603
	4	0.261	0.334	0.606
	Avg	0.259	0.337	0.608
Heat Meter W (150 F Mean)	1	0.262	0.346	0.643
	2	0.261	0.335	0.632
	3	0.261	0.337	0.614
	4	0.261	0.338	0.615
	Avg	0.261	0.339	0.626
Average conductivity at 150 F mean		0.260	0.338	0.617
Standard deviation		0.002	0.005	0.012
Conductivity as determined in a guarded hot plate apparatus (interpolated)		0.296	0.390	0.672
Heat Meter T (set at 350 F mean, normally at 500 F mean)	1	0.392		0.690
	2	0.380	0.403	0.684
	3	0.375	0.401	0.673
	4	0.376	0.404	0.685
	5		0.401	
	Avg	0.381	0.402	0.683
Heat Meter U (350 F mean)	1	0.405	0.414	
	2	0.397	0.415	0.705
	3	0.400	0.410	0.699
	4	0.395	0.410	0.710
	5			0.704
	Avg	0.399	0.412	0.705
Average conductivity at 350 F mean		0.390	0.407	0.694
Standard deviation		0.006	0.002	0.006
Conductivity as determined in a guarded hot plate apparatus (interpolated)		0.415	0.462	0.733

number of tests were performed and the distribution of results normal.) These results show that the reproducibility which the heat meter units are capable of producing consistently is well within the range necessary for the tests being carried on.

The first series of tests (Table 1) was made by an operator familiar with the use of heat meters for rapid determination of thermal conductivity. A second operator experienced in the use of a potentiometer for reading millivoltages but entirely unfamiliar with the heat meter apparatus ran a second series of tests after a limited training period. Conductivities obtained by Operator II and comparisons between these conductivities and those recorded by Operator I are given in Table 2.

The close agreement between the conductivities, as determined by these two operators, is further evidence of the generally good reproducibility of results possible with this apparatus for the samples tested.

The somewhat poorer reproducibility of conductivity results for sample C as compared with the other materials tested is believed the result of several factors. During this series of tests the sample shrank from 0.94 to 0.91 in., and exposure to high temperatures caused it to warp. This was especially noticeable following tests in the heat meter units (T and U) which were run with hot-surface temperatures about 500 F. Because of the hardness, strength, and warping characteristics of this material, the sample under test made poor thermal contact with the hot and cold surfaces of the heat meter units compared with mineral wool, for example. The amount of warping and degree of success in making good thermal contact could cause the observed variation in conductivity results.

TYPICAL APPLICATIONS OF THE HEAT METER APPARATUS

The heat meter apparatus for rapid thermal conductivity determinations is of good precision but not necessarily of good accuracy. As such it is ideal for use in comparison studies in groups of samples where information is desired on the effect of several variables on conductivity. It should not be used on a single sample of material to obtain *the conductivity* of this material. In other words, the heat meter apparatus will give a precise relative conductivity value for a particular sample which must be related to other samples of a similar nature or to a standard.

The use of the apparatus can best be illustrated by citing in detail five actual examples of problems where the heat meter apparatus has been used successfully.

1. *Moderate Temperature Block Insulation:* The problem was to determine if the conductivity of an insulation modified to improve a specific physical property was as low as that of the regular insulation. The heat meter apparatus is ideally suited for making this type of comparison. On successive days the same pilot plant equipment was used to make both the regular insulation and the modified insulation. Four samples of each type were tested in the heat meter apparatus at two mean temperatures each. The results of these tests are in Table 3.

The data show even without a detailed statistical analysis that the conductivity of the modified insulation was substantially that of the regular material. The small difference in results is easily accounted for by the slight difference in average density. Total test time required for 16 tests: two days on two of the four units.

2. *Friction Material:* Information was desired on the change in thermal conductivity of a friction material when it was impregnated and brass wire was added. With

these materials high conductivity is desired in order to improve the heat dissipation from the friction surface. Four sets of samples were prepared, impregnated and unimpregnated, with and without brass wire. As the thickness of this friction material was ⅛ in., it was necessary to use 8 layers to obtain the required 1 in. test thickness.

As shown in Table 4, both impregnation and the addition of brass wire increased the conductivity, the impregnation being about twice as effective. Total time required: one day on one of the four units.

3. *Experimental Block Insulation:* This was a new material with little information available on the effect that variables in composition had on thermal conductivity. It was decided that a Graeco-Latin square type of statistically designed experiment would

TABLE 2—REPRODUCIBILITY OF CONDUCTIVITY RESULTS—OPERATOR II

	THERMAL CONDUCTIVITY BTU/(HR) (SQ FT) (F DEG/IN.)		
	Sample		
	A	B	C
Heat Meter V	0.264	0.337	0.604
(150 F mean)	0.261	0.335	0.632
Heat Meter W	0.262	0.334	0.603
(150 F mean)	0.251	0.333	0.608
Operator II average	0.260	0.335	0.612
Operator I	0.260 ± 0.002	0.338 ± 0.005	0.617 ± 0.012
Heat Meter U	0.398	0.420	0.703
(350 F mean)	0.406	0.418	0.712
Operator II average	0.402	0.419	0.708
Operator I	0.399 ± 0.006	0.412 ± 0.002	0.705 ± 0.006

TABLE 3—MODERATE-TEMPERATURE BLOCK INSULATION COMPARISON OF THERMAL CONDUCTIVITY

SAMPLE		DENSITY, LB/CU FT	THERMAL CONDUCTIVITY BTU/(HR) (SQ FT) (F DEG/IN.)	
			At 150 F mean	At 350 F mean
Regular	1	10.4	0.34	0.42
	2	11.1	0.35	0.44
	3	10.1	0.33	0.42
	4	11.3	0.36	0.44
	Avg	10.7	0.34	0.43
Modified	1	10.1	0.33	0.42
	2	10.5	0.33	0.42
	3	10.5	0.34	0.43
	4	10.5	0.33	0.42
	Avg	10.4	0.33	0.42

TABLE 4—FRICTION MATERIAL—COMPARISON OF THERMAL CONDUCTIVITY

	THERMAL CONDUCTIVITY BTU/(HR) (SQ FT) (F DEG/IN.)		
	Wire	No Wire	Avg
Impregnated	2.5	2.1	2.3
Unimpregnated	1.6	1.5	1.6
	Avg 2.1	1.8	

yield the most data. Preliminary work showed that four variables, three factors in composition and density, would probably have the most effect on conductivity. Five widely spaced levels of each of these variables were studied in a 25 sample square. The composition of each individual sample and its density were varied in a set pattern so that each level of each variable occurred in combination with every other level of the other variables only once. Incidentally, following conductivity tests the samples were broken in a strength machine so that the effect of composition on both conductivity and strength were studied statistically with a single set of samples.

Average conductivity values with respect to each variable were computed. Since each individual average was taken over five tests the results are reliable. This means that the results of all 25 tests were used in studying each of the four variables. Because these same 25 tests were used four times, this is a true economy of both sample preparation effort and testing effort.

The results of this test program revealed how to make the best compromise between conductivity and strength based on the information available. Even more important, they pointed to the areas where further research was needed. Time required for conductivity test: 6¼ days on one of the four units.

4. *Rigid Fibrous Insulation:* Information was needed on the effect of heat flow direction on thermal conductivity in a rigid fibruous insulation. Strips of the material were cut 7/16 × 7/16 × 9 in. These strips were laid together side by side and the conductivity determined both with the heat flow normal to the original face, and with the strips rotated 90 deg, heat flow normal to the lengthwise edge. These same tests were also made in a guarded hot plate apparatus.

The results of these tests in Table 5 showed the increase in conductivity with the shift of heat flow direction. Because of cracks in the samples the check between the two apparatus is not exact; however, since only qualitative results were required the heat meter results were adequate. Test time required for the hot plate tests on two samples with one apparatus: 15 working days. Test time required for heat meter tests including one replication: One day on one of the four units.

In another comparison of the effect of heat flow direction on thermal conductivity, a group of four samples which had been tested earlier was retested in the heat meter apparatus. While not agreeing exactly in absolute values the results from two methods of test were in the same order so that the same conclusions would be drawn from either set of data. Test time required to make a series of determinations on four samples with one hot plate apparatus: 38 working days. Test time required for heat meter test on four samples: One day on one of the four units.

5. *Rigid Panel Insulation:* The effect on thermal conductivity of the three variables of amount of fiber, fiber length, and source of one of the constituents was needed. A factorial type of experiment was designed in which two levels of each of the three variables were investigated. This required eight different kinds of samples. Four replications were made for a total of 32 samples. All these samples were tested in the heat meter apparatus at 150 F mean temperature. As the experiment was originally planned the density of the samples was to be held approximately constant, the only variables being the three previously mentioned. A simple analysis of variance

TABLE 5—RIGID FIBROUS INSULATION—COMPARISON OF THERMAL CONDUCTIVITY

	THERMAL CONDUCTIVITY BTU/(HR) (SQ FT) (F DEG/IN.)	
	Heat Meter at 150 F mean	Guarded Hot Plate at 100 F mean
Heat flow normal to face	0.38	0.41
Heat flow normal to lengthwise edge	0.61	0.70

would then have shown the true significance of any conductivity differences observed. Since the samples were not all prepared at exactly the same density, the extremes being 30.9 lb per cu ft and 40.4 lb per cu ft, it was necessary to take density into account as a variable in the analysis. It was obvious from an inspection of the data that the amount of fiber was the most important variable of the three. Accordingly, the conductivity data were divided into two groups according to the amount of fiber in the samples. A regression of conductivity *vs.* density was calculated for each of these groups of 16 samples. The difference in the regression lines was caused by the change in fiber content. The effects of the other two variables were studied by the individual deviations above and below the regression lines.

Because of the large number of tests made, the conclusions drawn in the foregoing program were very reliable. Test time required: Four days with two of the four units.

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DISCUSSION

F. A. JOY, State College, Pa. (WRITTEN): The authors have shown large time savings by the use of heat flow meters and have obtained consistent and satisfactory results for the objectives outlined. In many other ways heat flow meters provide convenience and often the only way to get required information. Their use increases in spite of a widely shared suspicion of their absolute accuracy.

The causes of inaccuracy merit more study, and the authors' interpretation of their results—about 10 percent lower than the *A.S.T.M.* guarded hot-plate—would be very much valued.

Intrinsically, there is no reason why a properly designed heat flow meter should not be accurate. Of course, it must be properly calibrated and properly applied as well. Many observers have reported that the meter response depends on the surface

it contacts and that it must be calibrated *in place* or in a similar thermal environment. Some have noted the need for firm pressure or thorough sealing to the surface where applied. These precautions appear necessary in some designs, especially when the meter encourages a non-uniform heat flux by metallic paths through the thermal resistance in its core or when it has no guard area surrounding the area covered by thermocouples.

I would like to ask more about the calibration method used by the authors, in particular whether both faces of the meter were on metal. This arrangement might allow a larger proportion of heat to funnel through the center, due to through metallic paths, than would occur later in usage with only one metallic contact. We prefer to calibrate such meters between twin boards previously tested together in a guarded hot plate, adding a metal contact surface if the meter must be so used.

I am sure that heat flow meters are going to be more widely used and improvements in their design will accompany the demand.

L. F. SCHUTRUM, Cleveland, Ohio: I have a question concerning the positioning of the upper plate with respect to the sample. Was the positioning of the movable plate for a given sample a matter of a fixed dimension measured between plates, a given pressure on the sample, or was it arbitrary and left up to the judgment of the operator?

AUSTIN WHILLIER, Cambridge, Mass.: The authors are to be congratulated on developing a piece of equipment that should find wide use in many industrial fields. I should like to raise two questions in connection with its use.

The first is about the calibration of the thermocouple that is used to measure the temperature difference across the sample. I believe that, from the original calibration data of the thermocouple material, a curve could be constructed showing temperature as a function of *emf*. In fact a series of curves at different temperature levels would be needed since the thermoelectric power is a function of temperature as well as of temperature difference.

The second is in regard to the possible effect of fluctuations in the temperature of the hot or cold plate on the accuracy of the heat-meter reading. It appears from the diagrams that the heat meter rests directly on one of these plates. If there should be a small change in temperature of the plate against which the heat meter rests, say of the order of a tenth of a degree, the effect on the instantaneous heat meter reading will be quite significant. However, if a thin piece of material, *i.e.*, a 1/10 in. thick piece of synthetic resin, were inserted between the plate and the heat meter the damping of these small temperature fluctuations would almost eliminate the deleterious effects on the accuracy of the reading of the heat meter.

R. J. MINDAK, Chicago, Ill.: This apparatus would be very useful in comparing large numbers of samples but I would like to point out a limitation to its usefulness in its present design. Most of us concerned with insulating materials are interested in samples of thicknesses which are normally used in building construction, *e.g.*, batts 3½ to 4 in. in thickness.

An apparatus of this type will not give a very accurate measure of the heat flow through a thick sample because the heat flow is measured across one surface only in a relatively short time period. The thermal lag between the two surfaces induced by the heat storage capacity of the sample will have a marked effect on the average heat flow through the sample during the test period which is of course a condition of unsteady state heat transfer.

I think that the usefulness of this apparatus could be increased considerably by installing an additional heat flow meter on the other side of the sample and reading the heat flow across each surface. Experimentation will determine the testing time required to achieve the desired accuracy. It is felt that the 10 percent deviation from the guarded hot plate method could be reduced considerably by this simple addition.

It is hoped that these remarks may help to improve the usefulness and accuracy of the apparatus.

AUTHORS' CLOSURE (J. D. Verschoor): Professor Joy asks about the reasons for the 10 percent difference in conductivity between the heat meter apparatus and the *A.S.T.M.* guarded hot plate. We have investigated a number of factors which could be causing the discrepancy, but have never been able to find anything except very minor factors as the thermocouple emf to temperature difference conversion.

In the calibration of the heat meters, an attempt was made to duplicate as closely as possible the heat flow conditions that exist when the units are under operation. A guarded hot plate operated at 210-215 F was substituted for the lower steam plate in the heat meter apparatus. A heat meter to be calibrated was placed on each face of the hot plate, a piece of synthetic resin of the same thickness as the heat meter surrounded it, and finally a sample of insulation was inserted between each heat meter and cold plate. Each side of the guarded hot plate apparatus thus simulated the thermal conditions of the heat meter conductivity apparatus. Of course, it was also necessary to calibrate with the positions of the heat meters reversed to account for differences in the pieces, and then average the two calibrations.

The positioning of the upper assembly questioned by Mr. Schutrum was determined by the thickness of the sample when testing non-compressible samples. For compressible samples, spacer blocks were placed in the outside area of the sample. The weight of the upper plate plus its insulation and cover insured good thermal contact between the plates and the sample.

Dr. Whillier mentioned the thermocouple calibration. A nomograph could have been devised to arrive at an exact emf to temperature difference conversion, but to do so would have cut down on one of the factors of simplicity while achieving only a relatively unimportant increase in accuracy.

It is true that the instantaneous rate of heat transfer is greatly influenced by small changes in temperature of the surface adjacent to the heat meter. It was for this reason and ease of calibration that the heat meter was placed in contact with the steam plate rather than in contact with the tap water plate or the controlled, electrically heated plate. Before we insured that the steam was at a precise constant pressure and so at a constant condensation temperature, there were large fluctuations in heat flow due to the action of the steam traps. By making sure that the steam was at atmospheric pressure, the steam plates maintained constant temperature and the instantaneous rate of heat flow remained stable.

Mr. Mindak asks about testing 3½ to 4 in. samples. This would require a larger sample area than is available in the present apparatus for effective guarding. More sensitive heat meters would be needed to measure the reduced heat flow due to the diminished temperature gradient with the thicker samples. Added testing time would also be necessary as steady state heat flow could not be established in a period of 1½ hr with 4-in. samples. Furthermore, the conductance of a four inch sample may be calculated with sufficient accuracy from the conductivity of a one inch sample.

A second heat meter placed on the other side of the sample is not warranted in the present apparatus. The single heat meter measures the heat flow in only the center portion of the sample, the outer portions of the sample forming a guard area. Therefore, the quantity of heat going from the hot plate into the test area should equal the heat flowing from the test area into the sink. The second heat meter would give only a second measurement of the same heat flow.

A practical difficulty in installing a second heat meter is the fluctuations in the temperature of the adjacent surfaces which would affect the instantaneous rate of heat transfer. With tap water cooled plates, temperature variations are inevitable. Furthermore, the temperatures of the electrically heated plates are above the limit of the synthetic resin used in the heat meters.

1510

No. 1510

IRRADIATION OF VERTICAL AND HORIZONTAL SURFACES BY DIFFUSE SOLAR RADIATION FROM CLOUDLESS SKIES

By G. V. PARMELEE*, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

RECENT YEARS have seen a growing interest in solar radiation and its influence on engineering problems for several reasons. For example, the design of systems which utilize solar energy for heating in cold weather or for operating absorption refrigeration systems in hot weather requires detailed knowledge of solar energy intensities. Correlation of weather data to establish suitable bases for estimating cooling and heating loads must take solar radiation into account. Also the solution of the problem of improving the livability of buildings in hot climates without the benefit of air conditioning depends in part upon adequate solar intensity data. Many data are available, most of them as measurements of total solar radiation on a horizontal surface. These measurements are now regularly made at about 70 stations in the United States, its outlying possessions and in Canada. However, the data needed by the engineer are the components of the total incident solar radiation: (1) the direct or beamed radiation as a function of the altitude of the sun; and (2) the amount of diffuse radiation which falls on a surface of given orientation at a given time. The diffuse component arises from the scattering of the direct beam by various constituents of the atmosphere and on a vertical surface includes ground reflected solar radiation.

Considerable information on these two components of solar energy has been collected during the course of research[1, 2] carried on at the A.S.H.V.E. Research Laboratory in Cleveland to determine solar heat gain from various types of fenestrations. This research was carried on under the direction of the TAC on Heat Flow Through Glass†. Preliminary analysis of the data showed that the intensity of the direct beam varied widely for a given solar altitude and cloudless skies, and that as the intensity of the direct beam decreased, the intensity of the scattered or diffuse solar radiation tended to increase. This relationship has been developed and is the subject of this paper. It provides means for com-

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[1] Exponent numerals refer to References.

† Personnel: R. A. Miller, Chairman; N. B. Hutcheon, Vice Chairman; W. J. Arner, F. L. Bishop, E. W. Conover, D. D'Eustachio, W. B. Ewing, J. E. Frazier, J. S. Herbert, R. W. McKinley, E. C. Miles, D. R. Muir, H. B. Vincent.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1954.

puting the direct and diffuse components of the total radiation which falls on a horizontal plane.

A second step permits an estimate to be made of the diffuse and direct components of the radiation which falls on a vertical surface of any orientation. This approach differs from those made by others, who, (1) give mean hourly, daily or seasonal values of total radiation falling on various surfaces, mean ratios of direct or total to diffuse radiation, or mean values of diffuse radiation for

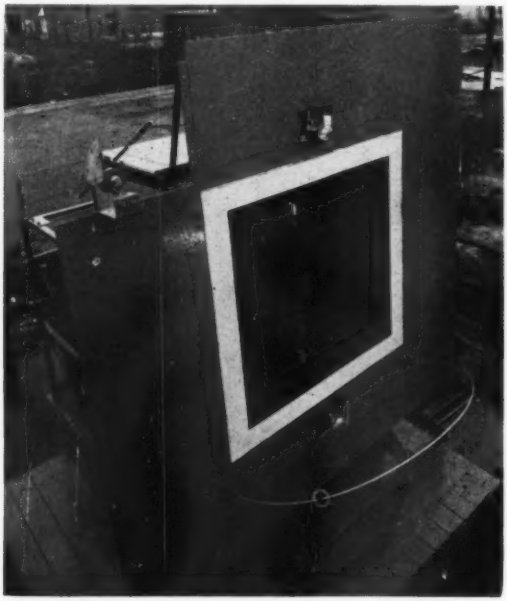


FIG. 1. CLOSE-UP VIEW SHOWING PYRHELIOMETERS MOUNTED ON CALORIMETER FOR MEASURING SOLAR RADIATION ON A VERTICAL SURFACE. A SHADING DISK AND THE METHOD OF ADJUSTING IT IS ALSO SHOWN. THE CALORIMETER FACES WEST

cloudless skies,[3-8] or (2) give methods of computing these values in terms of atmospheric conditions.[9] Additional investigation is needed to test and refine the generality of the relationships presented here and it is hoped that the present approach will prompt such study.

SOURCE OF DATA

Period and Methods of Observation: Observations of the direct and diffuse components of solar radiation on a horizontal plane have been made throughout the course of all of the research on heat flow through glass reported in the references mentioned. In addition, many tests were made in which the special calorimeter (see Fig. 1) used in the research was set so that the test section

was vertical with a fixed orientation. In all of these tests, measurements were also made of the direct and diffuse components of the solar radiation incident upon this vertical surface. In addition to data so gathered, several series of special observations were made in which the calorimeter apparatus was rotated through a full circle during which operation observations were made of the diffuse solar radiation incident upon its vertical surface at intervals of 20 to 30 deg of rotation. Measurements on a horizontal surface were made simultaneously. Most of the observations were made during the months of April through



FIG. 2. SOLAR RADIATION ON A HORIZONTAL SURFACE WAS MEASURED BY A PYRHELIOMETER MOUNTED ON THE TOWER. THE TOWER IS ON THE NORTHWEST CORNER

September between 1947 and 1952, but data given in this paper include measurements made in all the remaining months of the year except January.

Sky Conditions: This report includes data obtained on 30 days during this period which were wholly cloud-free, both as observed at the Laboratory and as reported at the Cleveland Weather Bureau, about 10 miles to the southwest. In addition, values are included for portions of days in which the sky was observed to be free of clouds. Despite this cloud-free condition, the clearness of the sky varied greatly from very clear days, usually those with relatively low dew-points and low dry bulb temperatures, to very hazy days produced by the combination of large amounts of atmospheric water vapor with industrial contamination. Because sky conditions change quite rapidly, they were observed and recorded at 15 min intervals.

Ground Reflected Solar Radiation: The irradiation of vertical surfaces depends not only upon the direct and diffuse solar radiation received from the sky, but

also upon solar radiation reflected from the ground. In general, the ground reflectance was low, as the surroundings consisted of dark areas of which the backgrounds of Fig. 1 (camera looking southeast) and of Fig. 2 (camera looking northeast) are typical. The immediate foreground was the roof of the test house with a measured light reflectance of about 5 percent. The average reflectance of all the surroundings is estimated to be about 10 percent. The horizon was cut off in some directions by nearby buildings with a maximum altitude of about 12 deg for the horizontal surface (the top of the tower shown in Fig. 2) and up to 15 deg with respect to the centerline of the vertical surface of the panel.

Instrumentation: All observations were made with pyrheliometers which have a *hemispherical vision.* Two were mounted on the calorimeter as shown in Fig.

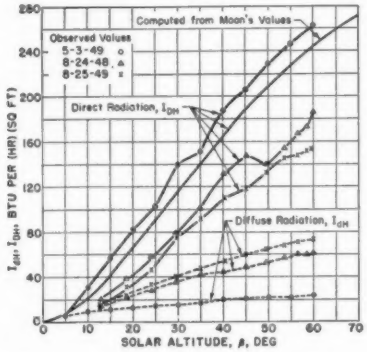


FIG. 3. TYPICAL VALUES OF SOLAR RADIATION ON A HORIZONTAL PLANE FOR CLOUDLESS SKIES OF DIFFERENT DEGREES OF CLEARNESS

1 and a third was mounted horizontally on the tower at the northwest corner of the test house roof (shown covered in Fig. 2). Measurements of diffuse radiation were made by shading the thermopile element from the direct rays of the sun by means of shading disks of such diameter and distance from the element as to exclude a 5 deg 43 min cone in the direction of the sun. One is shown in Fig. 1. Readings of electromotive force (emf) were recorded mainly on a 0-5 millivolt range electronic recording potentiometer. Total radiation on the horizontal surface was recorded at least once every 5 min and diffuse radiation recorded once every 30 to 45 min. Most vertical surface data are averages of from four to eight pyrheliometer readings over a period of 7½ to 15 min.

The constants of the various pyrheliometers for diffuse solar radiation require some comment. In all, four different pyrheliometers were used. Comparisons under identical conditions showed that one instrument, No. 1979, developed about 15 percent more emf when exposed only to diffuse radiation on the clearest of days than the other three instruments, although all four gave the same values on overcast days and also the same values for direct beamed radiation. To make the data consistent, a sliding scale of reductions was applied to the readings of

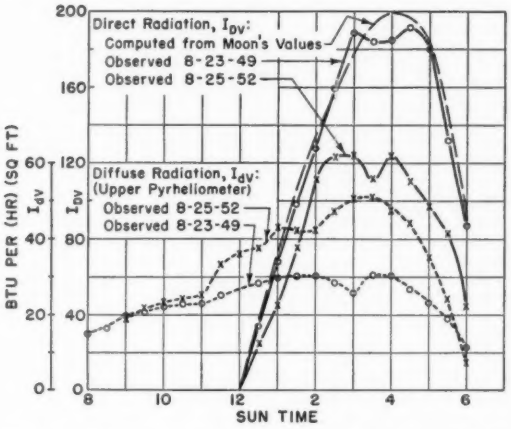


FIG. 4. TYPICAL VALUES OF SOLAR RADIATION ON A WEST FACING VERTICAL SURFACE FOR CLOUDLESS SKIES OF DIFFERENT DEGREES OF CLEARNESS

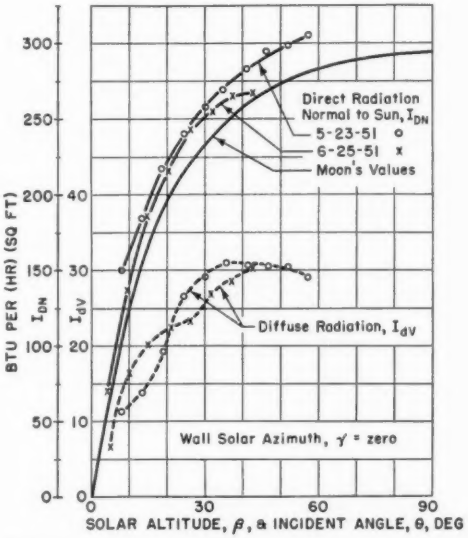


FIG. 5. VARIATION OF VERTICAL SURFACE DIFFUSE RADIATION WITH SOLAR ALTITUDE

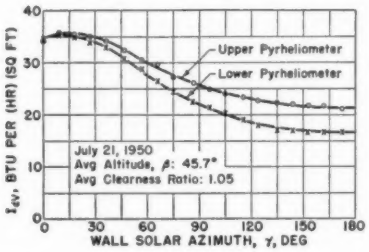


FIG. 6. VARIATION OF VERTICAL SURFACE DIFFUSE RADIATION WITH ORIENTATION

instrument No. 1979. A reduction of 15 percent was applied when the sky was clearest and no reduction was applied when the sky was very hazy.

TYPICAL OBSERVATIONS

Figs. 3, 4, 5 and 6 give typical observations of direct and diffuse solar radiation for cloudless skies with one or more of the variables held essentially constant in each case represented. Fig. 3 gives values of radiation received on a horizontal plane for three different degrees of atmospheric clearness. Fig. 4 gives values of radiation received by a vertical surface with fixed orientation (west) for two different degrees of atmospheric clearness. To compare these data with those for an atmosphere of specified atmospheric conditions, direct radiation values from Moon's[10] work are shown. His values have been used as a basis of comparison throughout this paper and the ratio of the observed intensity of direct radiation to Moon's value for the same solar altitude has been designated the clearness ratio. Figs. 3 and 4 show that for the same solar altitude, the intensity of the diffuse radiation incident upon a given surface decreases as the intensity of the direct beam increases. The various angles referring to sun and wall positions in this paper are described by Fig. 7.

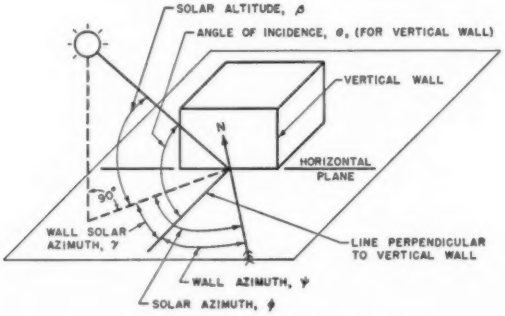


FIG. 7. DEFINITION OF SOLAR ANGLES

The influence of solar altitude, β, on vertical surface diffuse radiation with constant wall solar azimuth, γ, and very nearly constant clearness ratio is shown by Fig. 5. The influence of orientation on vertical surface radiation values for nearly constant altitude and clearness ratio is illustrated by Fig. 6. This also shows the difference in the values registered by the upper and lower pyrheliometers.

ANALYSIS OF DATA

Horizontal Surface Values: Values of direct and diffuse solar radiation intensities for cloudless skies were computed from chart records at 10-deg increments of solar altitude and plotted as shown in Fig. 8 with solar altitude as the parameter. The straight line curves were fitted to each set of points by the method of least squares. The equations for these curves have the form

$$I_{dH} = X - Y\,(I_{DH}),\ \text{Btu per (hr) (sq ft)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

I_{dH} = diffuse solar radiation incident on a horizontal plane, Btu per (hour) (square foot).
I_{DH} = direct solar radiation incident on a horizontal plane, Btu per (hour) (square foot).

The constants X and Y are given in Table 1 and have been extrapolated as shown in Fig. 9 to obtain values for solar altitudes of 10, 70, 80, and 90 deg. Extrapolated curves for these altitudes are shown in Fig. 8 as dotted lines. The

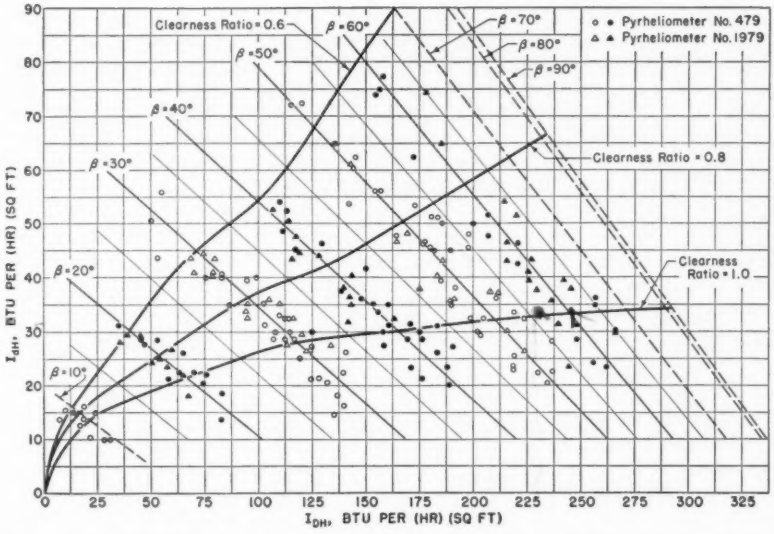


FIG. 8. DIFFUSE SOLAR RADIATION, I_{dH}, ON A HORIZONTAL SURFACE *vs.* DIRECT RADIATION, I_{DH}, ON A HORIZONTAL SURFACE AND SOLAR ALTITUDE, β

few data available for an altitude of 10 deg are shown for comparison. To show the relationship of these data to the clearness of the atmosphere, lines of constant clearness ratio have been drawn. Intersections of these curves with the straight line curves give the direct radiation on a horizontal plane and the corresponding diffuse radiation value for a given altitude of the sun. Light lines are for intermediate altitudes.

Two useful relationships derived from the straight line curves of Fig. 8 are shown in Figs. 10 and 11. Fig. 10 is a family of curves which shows the ratio

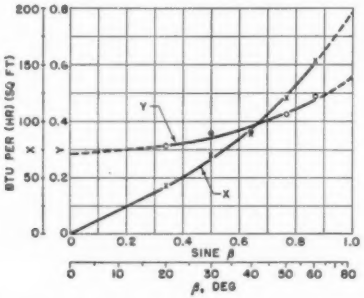


FIG. 9. CONSTANTS X AND Y *vs.* SINE OF SOLAR ALTITUDE, β

of the total to diffuse solar radiation on a horizontal plane as a function of solar altitude and the total solar radiation on a horizontal plane. With these curves, the direct and diffuse components can be determined for any measurement of total radiation on a horizontal plane, if the solar altitude is determined for the time of the measurement. Total radiation is the usual measurement at Weather Bureau Stations. Fig. 11 shows this same ratio plotted against direct radiation on a plane normal to the sun's rays, I_{DN}, a measurement made at some stations. The curve was computed from the equations listed in Table 1 and gives the diffuse radiation with an accuracy of about 5 percent.

TABLE 1—CONSTANTS[a] FOR DIFFUSE RADIATION ON A HORIZONTAL PLANE FOR THE EQUATION:

$$I_{dH} = X - YI_{DH}$$

COLUMN A SOLAR ALTITUDE β, DEG	COLUMN B X	COLUMN C Y	COLUMN D I_{DN} WHEN $I_{dH} = 0$
10	(20)	(0.295)	(391)
20	42.7	0.314	(397)
30	70.4	0.360	(391)
40	90.1	0.362	(387)
50	121.4	0.424	(374)
60	153.6	0.492	(360)
70	(175)	(0.520)	(358)
80	(192)	(0.545)	(358)
90	(198)	(0.560)	(354)

[a] Values in parentheses determined by extrapolation.

Vertical Surface Values: Vertical surfaces introduce another parameter in addition to solar altitude and atmospheric clearness, namely the wall solar azimuth angle, γ (see Fig. 7). Vertical surface values for cloudless skies were plotted with clearness ratio as abscissa for altitude intervals of 10 deg and azimuth intervals of 10 deg up to $\gamma = 75$ deg and for larger intervals at greater values of γ. Curve families for each altitude interval were drawn. Fig. 12 for the 40 to 50 deg interval of solar altitude is typical. Because of the large num-

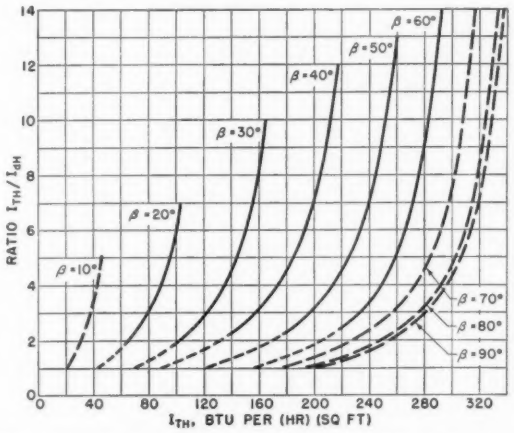


FIG. 10. RATIO OF TOTAL TO DIFFUSE SOLAR RADIATION ON A HORIZONTAL SURFACE *vs.* TOTAL RADIATION, I_{TH}, ON A HORIZONTAL SURFACE AND SOLAR ALTITUDE, β

ber of variables to be covered, considerable interpolation was necessary despite the fairly extensive data collected.

From these curves and cross plots of these curves, curve families of vertical surface radiation, I_{dV}, were plotted against altitude for constant clearness ratios as shown in Fig. 13. With the aid of data of the type illustrated by Fig. 5, these curves were extrapolated to an altitude of 90 deg, for which vertical surface radiation would be the same for all orientations.

DISCUSSION OF RESULTS

Atmospheric Absorption and Scattering: The foregoing data show that a fairly well defined relationship exists between the clearness of the atmosphere and the direct and diffuse components of solar radiation for a given solar altitude and surface orientation. The atmospheric clearness or turbidity has been defined in this paper as the ratio of the observed intensity of the direct beam of solar radiation on a plane perpendicular to the rays to the value[10] for a standardized atmosphere for the same altitude of the sun. This ratio has the great merit of simplicity and definiteness in describing the condition of the

cloudless sky. It does not, however, take into account whether the direct beam is depleted by absorption or by scattering. This point will now be examined.

Depletion of the direct beam in a cloud-free sky is due to absorption and scattering. Ozone in the upper atmosphere and water vapor are the principal absorbing media. In all probability solid particles suspended in the air absorb to some extent. These elements plus the air molecules themselves scatter solar radiation. If it be assumed that atmospheric dust scatters but does not absorb, it is apparent that the intensity of diffuse radiation, directly proportional to scattering but reduced by absorption, can be expected to vary for the same intensity of the direct beam. The data shown in Fig. 8 have not been examined with this in mind, but it seems likely that the manner in which the direct beam has been depleted is responsible in part for the scatter of the data.

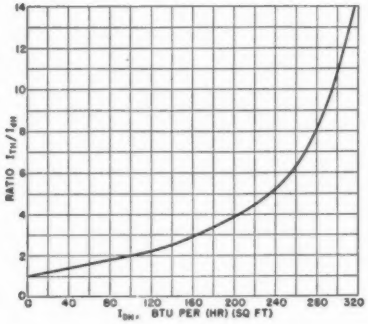


FIG. 11. RATIO OF TOTAL TO DIFFUSE SOLAR RADIATION ON A HORIZONTAL SURFACE *vs.* DIRECT RADIATION, I_{DN}, ON A SURFACE NORMAL TO THE SUN'S RAYS

Sources of Error: A certain amount of scatter can be attributed to experimental error. Some errors, however, such as improper shading of the pyrheliometer to obtain diffuse values, and chart reading errors, particularly at low intensities of diffuse radiation, are minimized in plotting. If the diffuse value is too high, then the direct radiation values, which are obtained from the difference between the total and diffuse emfs, will be too low. Hence, the point is simply shifted along the curve without much change in relative position. Other factors which affect the relative position of the plotted points are the presence of unreported clouds, abnormal reflection effects from surroundings. A third pertains mainly to horizontal surface values. To obtain these values for a given solar altitude, the diffuse radiation value usually required interpolation in the chart record. The value so obtained may not always have been typical of the sky conditions at the time the total radiation value was observed.

Horizontal Surface Observation: Except where the solar altitude β equals zero, a value of zero diffuse solar radiation represents a condition of no atmosphere. Therefore, one would expect the straight line curves of Fig. 8, when extrapolated to $I_{dH} = 0$ would give values of the direct radiation, I_{DH}, which.

when divided by the sine of the altitude, should yield the solar constant. According to Moon[10], the solar constant is 419 Btu per (hr) (sq ft) for a plane normal to the sun's rays when the earth is at its mean distance from the sun. The Smithsonian scale of pyrheliometry puts the constant about 3 percent higher or 429. A value of 410 is obtained when Moon's constant is corrected for the mean distance corresponding to the period represented by the bulk of the data. The extrapolated values are listed in Table 1.

The fact that the extrapolated values are of the same order of magnitude as the solar constant suggests that the mean curves represent a relationship that is general and hence applicable with reasonable accuracy to other locations. It can

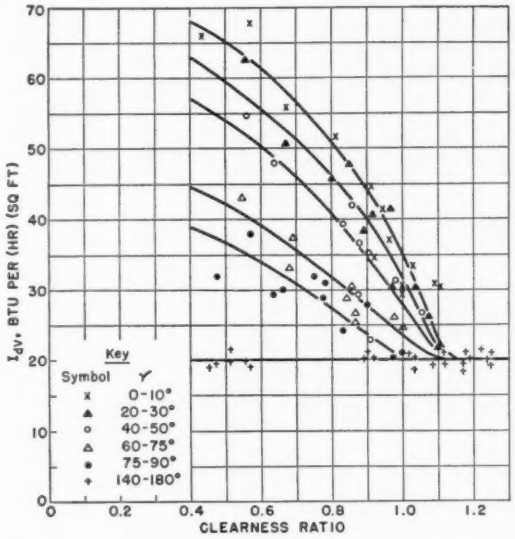


FIG. 12. DIFFUSE SOLAR RADIATION, I_{dV}, ON A VERTICAL SURFACE *vs.* CLEARNESS RATIO AND WALL SOLAR AZIMUTH, γ, FOR SOLAR ALTITUDES BETWEEN 40 AND 50 DEG

be shown that of the total radiation incident on a unit area at the center of a hemisphere of uniform radiant intensity, only 3 percent is contributed by the spherical zone bounded by altitudes between 0 and 10 deg. Hence, obstructions of the horizon at the point where these observations were made appear to be of minor importance.

The deviations from the solar constant of the values in Col. D of Table 1 for angles of 10, 70, 80 and 90 deg cannot be given much weight because they are extrapolations of extrapolations. The value for 60 deg is 12 to 15 percent low, depending upon which constant is taken for comparison.

It will be recalled that the data obtained with pyrheliometer No. 1979 were plotted after multiplying the diffuse values by a sliding factor which ranged from

0.85 for the clearest skies to 1.00 for clearness ratios less than 0.5. Because the pyrheliometer constants are based upon response for the entire solar spectrum whereas the shortest wave lengths predominate in clear sky diffuse radiation, it appears that No. 1979 is less sensitive to spectral variation in the incident radiation and hence may register diffuse solar radiation more accurately. These two points suggest that the slopes of the curves, especially at the higher values of β, should be decreased, that is, the No. 1979 values uncorrected would be more nearly correct.

Since there are insufficient data to be conclusive, no adjustment seems warranted at this time. Further study of the response of this type of instrument with respect to diffuse radiation seems desirable.

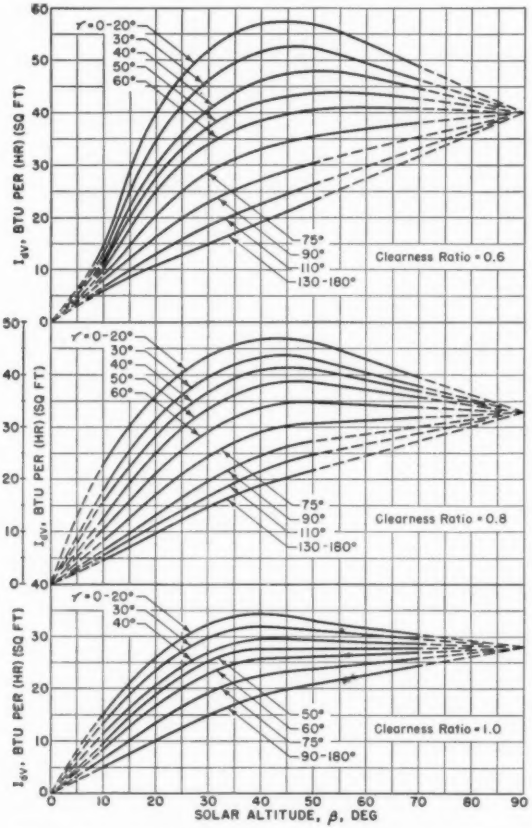


FIG. 13. DIFFUSE SOLAR RADIATION, I_{dV}, ON A VERTICAL SURFACE *vs.* SOLAR ALTITUDE, β, AND WALL SOLAR AZIMUTH, γ

A partially cloudy sky will greatly increase diffuse radiation values by reflection without impairing the intensity of the unobstructed direct rays of the sun. Curves similar to those of Fig. 8 may possibly be found to apply to skies of different degrees of cloudiness.

Vertical Surface Values: Interpretation of the vertical surface values requires consideration of three components of diffuse radiation, (1) ground reflected radiation, (2) radiation from the sky vault, and (3) radiation from that portion of the sky in the immediate vicinity of the sun. It is a matter of observation that the intensity of radiation from the sky is greatest near the sun, and that this increases as the clearness of the sky decreases. For a given clearness of the sky and solar altitude, the contribution of items 1 and 2 should be substantially the same for all orientations. At the point of observation, however, the terrain was not the same in all directions and for no orientation was the horizon wholly unobstructed. Hence, the data collected show some variation with orientation for the same value of γ.

The data of Fig. 6 show that with clearness ratio and solar altitude approximately constant, the vertical surface diffuse radiation decreases as the wall solar azimuth γ increases, slowly at first and then more rapidly. In this particular case, the component from the vicinity of the sun is small so that one would not expect the intensity to change much for angles greater than $\gamma = 90$ deg. The fact that I_{dv} does decrease continually until γ equals about 140 deg is probably due to the fact that, in this case, the effect of obstructions on the horizon increased as γ increased.

The curve family of Fig. 12 shows clearly the influence of component three, the radiation from the vicinity of the sun. The intensity of this component is proportional to the cosine of the incident angle, θ, and since $\cos \theta = \cos \beta \cos \gamma$, $\cos \theta$ decreases as γ increases. The figure shows that this component is substantially zero when the clearness ratio is about 1.1, that is, the sky is everywhere of uniform intensity, and that it increases as the clearness ratio decreases. Curves like those of Fig. 12 show, for γ greater than 130 deg, that I_{dv} is unaffected by clearness of the sky when β does not exceed about 45 deg. This is apparently due to the fact that as the sky vault component increases the ground reflected component decreases. Further, one would expect that for a given altitude, I_{dv} would be the same for all values of γ greater than 90 deg. The data show this is true for high clearness ratios and all values of β. For low clearness ratios, however, there are no data to show what happens when β is greater than 45 deg, but it has been assumed that the total value of I_{dv} for angles greater than $\gamma = 90$ deg increases as clearness ratio decreases. Hence, for clearness ratios of 0.6 and 0.8 estimated curves have been drawn in Fig. 13 for values of γ intermediate between 90 and 180 deg.

PRACTICAL APPLICATIONS

Within the limitations of present day solar radiation instrumentation and more extensive tests or refinements, the relationships developed in Fig. 8 for horizontal surfaces appear to have general practical value. Though the data as presented for vertical surfaces apply only to the specific location, they can be applied to other surroundings, first, by making a correction for different ground reflectance, and second, by making allowances for different horizon obstructions.

The correction for ground reflected radiation, I_G, is:

$$I_G = 0.5\,(I_{dH} + I_{DH})\,(\rho_e - 0.10),\ \text{Btu per (hr) (sq ft)} \quad . \quad . \quad . \quad . \quad (2)$$

where

ρ_e = the ground reflectance expressed as a decimal.

The correction for horizon obstruction can only be roughly approximated. It is estimated that the I_{dV} values for a surface viewing an unobstructed horizon would be greater by an amount equal to 0.15 times the diffuse values for γ greater than 90 deg for clearness ratio = 1.0. For clearness ratios of 0.6 and 0.8, the correction is 0.15 times the values for γ greater than 130 deg.

No allowance seems necessary for the small horizon obstruction in the case of the horizontal values presented. However, in some cases it may be found necessary to make some reduction in horizontal diffuse values because of partial shading of the sky by trees or tall buildings.

Diffuse values can be estimated if the solar altitude and either the total solar radiation on a horizontal plane, I_{TH}, or the direct normal radiation, I_{DN}, are known. If neither is known, the latter can be estimated from data given by Klein[9]. *Examples 1* and *2* will illustrate the use of the data.

Example 1: Find the diffuse solar radiation incident on a horizontal surface and on vertical surfaces facing east, south and west at 2 p.m. on August 1, 40 deg north latitude. The atmospheric clearness ratio is 1.0 and ground reflectance, $\rho_e = 0.20$; the horizon is unobstructed.

Solution: In Chapter 13, of THE GUIDE 1954, Table 6 gives the solar altitude, β, and wall solar azimuth, γ. For this example β is 56.0 deg. The values of γ as tabulated apply only to times when the sun strikes the vertical surface. Therefore, γ for the south and west walls at 2 p.m. is 59 deg and 31 deg respectively, while γ for the east wall at 2 p.m. is 180 deg minus the value for the west wall at the same time or 149 deg.

From Fig. 8, I_{dH} is found to be 33.0 Btu per (hr) (sq ft), corresponding to $I_{DH} = 231$. I_{DH} may also be found by multiplying 279, the value of I_{DN} in Table 4 of Chapter 13, THE GUIDE 1954 for 56 deg altitude and a clear atmosphere (Moon's values) by 0.829, the cosine (from Table 5, Chapter 13, THE GUIDE 1954) of the incident angle for a horizontal surface. The total radiation on a horizontal surface, I_{TH}, is 231 + 33 or 264 Btu per (hr) (sq ft).

The vertical surface values of diffuse radiation are found from Fig. 13 to be 22.0, 26.5, and 31.0 for east, south and west facing walls respectively. According to Equation 2, the correction for ground reflection is 0.5 (264) (0.20 − 0.10) or 13.2 Btu per (hr) (sq ft). The approximate correction for an unobstructed horizon is 0.15 × 22.0, or 3.3 Btu per hr, so that the corrected values are 38.5, 43.0 and 47.5, respectively.

Example 2: The total radiation on a horizontal plane when the solar altitude, β, was 60 deg and the sky was cloudless, was observed to be 260 Btu per (hr) (sq ft). Find the magnitude of the diffuse and direct components for the horizontal surface and the clearness ratio.

Solution: Fig. 10 shows that the ratio of total to diffuse radiation is 5.1. The diffuse component is therefore 260 ÷ 5.1 or very closely 51 Btu per (hr) (sq ft) and the direct component is 260 − 51 or 209 Btu per (hr) (sq ft). For a clearness ratio of 1.0 and $\beta = 60$ deg, I_{DH} is found from Fig. 8 to be 245 Btu per (hr) (sq ft). Therefore, in this example the clearness ratio is 209 ÷ 245 or 0.85. Diffuse radiation values for vertical surfaces can be found for this clearness ratio by the procedure illustrated

by *Example 1*. However, interpolation in Fig. 13 will be required. This illustrates how these procedures are applied to the usual observations of solar intensity made by the U. S. Weather Bureau.

CONCLUSIONS

1. Observations of solar radiation intensities on horizontal and on variously oriented vertical surfaces under conditions of cloudless skies have been analyzed. Relationships between the intensity of the direct and the diffuse components have been developed and expressed in terms of atmospheric clearness and position of the sun.

2. Based upon these observations, curves have been drawn which permit an estimate to be made of the diffuse solar radiation incident upon a horizontal surface or on a vertical surface of any orientation, if the solar altitude and either the total or the direct radiation on a horizontal surface are known. Methods of making corrections to the values for vertical surfaces for differences in ground reflectance and horizon obstruction are given.

3. Additional study of the relationships developed in this paper with data collected at other locations and under other conditions is suggested.

ACKNOWLEDGMENT

The assistance of many persons of the A.S.H.V.E. Laboratory staff in collecting and preparing the data is acknowledged with thanks. Special thanks are due to W. W. Aubele, former Research Engineer, A.S.H.V.E. Laboratory, who was associated with the writer during the collection of the major part of the data and some preliminary analyses.

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DISCUSSION

T. H. MACDONALD† and SIGMUND FRITZ,† Washington, D. C. (WRITTEN): This is an excellent analysis of the authors' data in a form which will be useful to many people. Our comments, mostly of a minor nature, are as follows:

1. Mr. MacDonald has found that measurements with Eppley pyrheliometers mounted in a vertical plane are likely to indicate values of radiation that are too low by 4 or 5 percent.

2. When the solar altitude is zero, there still is diffuse radiation which represents the beginning of twilight.

Qualitative reasoning suggests that linear extrapolation in Fig. 8 might not be quite correct in order to find the solar constant. It would be expected that the extrapolation would be curvilinear in such a way as to give larger values of the solar constant than those given in Table 1. The curvilinear effect would probably not be large, however.

The reason for the deviation of pyrheliometer No. 1979 is certainly not obvious; however, if spectral response is the cause, might it not be more likely that instrument No. 1979 is *more* sensitive to spectral variation since it differed from three other pyrheliometers?

I. F. Hand has published an analysis of diffuse radiation in the *Monthly Weather Review* for February 1954. In general the range of diffuse radiation values is not as great as the Cleveland values. A brief analysis of Mr. Hand's data indicates that it would corroborate the authors' Fig. 8 quite well for a solar altitude of 30 deg. For a solar altitude of 60 deg, Mr. Hand's data give about 20 percent higher values of diffuse radiation for a given direct radiation.

L. R. PHILLIPS, Hartford, Conn.: If the glass was considered perfectly transparent in these tests, were there any temperature measurements taken of the glass indicating its actual temperature rise?

In the diffuse component, was any note taken of the different types of building surfaces that would reflect the radiation, such as a close brick wall of an adjacent building and the possibility of a change in wave length causing a high component of lower wave length that would change the temperature of the glass?

T. F. MALONE,* Cambridge, Mass.: Meteorologists interested in the application of climatic data to problems of building and equipment design have followed with considerable interest the research on this topic conducted at the A.S.H.V.E. Research Laboratory.

If the generality of the results presented here can be confirmed, a step of some significance towards the establishment of a real radiation climatology will have been made.

Further examination of the effects of absorption by water vapor, which can be as much as 10 to 15 percent, appear to be in order. This would probably be of considerable help in affirming the validity of extrapolating the diffuse radiation to zero in order to determine the radiation incident on the outer limit of the atmosphere.

Cursory examination of the weather maps associated with the two extremes given in Fig. 3 suggested that there was a significant difference in the amount of the precipitable water on those two days.

Of greatest interest to me is the inviting possibility that the type of analysis presented here might be extended to the analysis of direct and diffuse radiation on other than cloudless days. From work that we have done at M.I.T. along this line, but from

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a slightly different approach, it would seem worth while to explore this further.

It is suggested that observations such as those made by Kimball and Hand in the *Monthly Weather Review*, Volume 50, pages 615 to 628, 1922, might provide an interesting check on the variation of diffuse radiation on vertical wall azimuth.

Has the author had an opportunity to check his results against the diffuse radiation measurements at Blue Hill Observatory?

With respect to the corrections for pyrheliometer No. 1979, it would seem that this instrument is more sensitive to spectral variations in incident radiation rather than less sensitive.

R. C. EDWARDS, Pompton Plains, N. J.: How does the total radiation for any given set of conditions vary with clearness ratio?

R. W. MCKINLEY, Pittsburgh, Pa.: I would like to ask the authors to indicate the source of the illustration used in the first slide. I think that it contributes to a general understanding of the various aspects of the problem.

In the paragraph preceding the section *Vertical Surface Values* the authors state that a partially cloudy sky will greatly influence diffuse radiation values by reflection. They suggest that there is a possibility for setting up some curves for this type of condition. I wonder whether or not the available Weater Bureau data might not in some way be utilized for this purpose, now that the clearness ratio concept has been presented.

AUSTIN WHILLIER: My interest in solar radiation is primarily in its use for house heating, water heating and power generation. An important item in the calculation of the performance of solar energy collectors is the diffuse solar radiation incident upon the collector surface. In the past it has always been necessary to make an educated guess as to the magnitude of diffuse radiation, and the problem has been all the more difficult because solar collectors in general are neither horizontal nor vertical, but tilted.

This correlation of diffuse radiation on both vertical and horizontal surfaces by Mr. Parmelee fills a big gap in our knowledge of solar radiation; it is a development that solar engineers have awaited for some years. From the data presented it is now possible to get a very good approximation of the diffuse radiation incident upon any tilted surface. This tilted diffuse radiation is *approximately* equal to the weighted sum of the horizontal and vertical diffuse radiation, weighted by the factors $(1-\theta/90)$ and $\theta/90$, respectively, where θ is the angle of tilt from the horizontal. The error in this approximation is not over 10 percent and probably less than 5 percent.

AUTHOR'S CLOSURE: In reply to Mr. Phillips' question on glass temperatures, I can say that we have reported some measurements in several research papers dealing with the problem of heat flow through glass. The reflection of solar radiation from nearby buildings is an interesting point, but we have made no attempts to determine how much diffuse radiation came from such sources. The light reflectance of the nearest surface, that of the test-house roof, is low, about 5 or 6 percent. The average reflectance of the terrain around the house is judged to be about 10 percent. The long-wave low-temperature radiation incident on our test sections was also measured, but again no attempt was made to determine the contribution of the various sources.

Professor Malone's comments are appreciated. I doubt if many of you are familiar with the excellent work done by him and his associates in analyzing diffuse radiation from cloudy skies. They present a much more difficult problem. This work was done as part of a research project for the *H.H.F.A.* in applying climatic data to house design.

Professor Malone suggests that differences in diffuse radiation for the same solar altitude are due to differences in the amount of water vapor in the atmosphere. This is probably the most important factor. Differences in diffuse radiation for the same intensity of the direct beam component at the same solar altitude may be caused by differences in the manner in which the direct beam is depleted in its passage through

the atmosphere. Because water vapor and dust have different effects, further study of these or other data on diffuse radiation might lead to useful information on atmospheric dust. I doubt, however, if our data are extensive enough for such a study.

We have not had much opportunity to compare our observations of diffuse radiation with those made at Blue Hill by Mr. Hand. In general his data for horizontal surfaces fall near our curves; but because they appear to be mainly for very clear days, they do not fully test the generality of our curves. We look forward to the publication of his observations for south-facing walls. I am not familiar with the 1922 paper of Kimball and Hand on this subject and appreciate the reference to it.

The question of sensitivity of pyrheliometer No. 1979 is perhaps a matter of interpretation of this term. My thought was that perhaps the spectral response of this instrument was about the same at all wave lengths whereas the response of the other instruments was less only in the shorter wave length region. On very clear days, when a large part of the diffuse radiation is in the short or blue part of the spectrum, this instrument would then give a greater output than the others. On hazy days, when much less energy is in the blue end, all would respond about the same. This is wholly conjecture, of course, and the real reason might be quite different.

Mr. Edwards has raised a very practical question: How does the total radiation received vary with clearness ratio? With a clearness ratio of about 0.7, the total radiation is roughly 20 percent less than on a day with clearness ratio equal to one. The reduction is less than the reduction in clearness ratio.

In answer to Mr. McKinley, the first slide was published in the 1948 A.S.H.V.E. TRANSACTIONS on p. 409 as Fig. 2. The clearness ratio is an attempt to describe general atmospheric conditions on a cloudless day. Actually it describes the atmosphere in the path of the direct beam. It is of doubtful value for cloudy days as it could apply only when the sun was not obscured by clouds. For the same clearness ratio, diffuse radiation on cloudy days would be greater than on cloudless days, because of reflection of direct-beam radiation by the clouds. Professor Malone and his group at M.I.T. have made an excellent study of Blue Hill cloudy day diffuse radiation.

It is gratifying to hear Dr. Whillier's comment that the data on diffuse radiation has application in estimating this component for inclined surfaces, such as those of solar energy collectors. We do have some data on diffuse radiation on inclined surfaces, but nothing that is very systematic.

1511

No. 1511

CHARACTERISTICS OF DOWNWARD JETS OF HEATED AIR FROM A VERTICAL DISCHARGE UNIT HEATER

By LINN HELANDER*, S. M. YEN**, AND L. B. KNEE†, MANHATTAN, KANS.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station of Kansas State College.

THIS PAPER is the third of a series on downwardly projected jets of heated air giving the results of research conducted under the guidance of the A.S.H.V.E. Technical Advisory Committee on Air Distribution‡. The previous work may be briefly summarized as follows:

The first paper[1] dealt with the results of a preliminary study of jets from two outlets, and presented a tentative equation for estimating the downward throw of jets of the type studied. Using symbols listed in Nomenclature, this equation has the form:

$$L_{max}/D_o = (1/a)\,[(a\,\overline{B}_o + 1)^{\frac{1}{2}} - 1] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

$\overline{B}_o$ is denoted as the cross-stream average buoyancy number at the outlet.

The second paper[2] dealt with the down throw of approximately free jets from long-radius *A.S.M.E.* nozzles attached to the base of a plenum chamber. In this second paper, an extensive amount of data on downward throws greater than approximately 8 orifice diameters and less than 45 orifice diameters were presented. These data were correlated by the empirical equation:

$$L_{max}/D_o = 1.66\,B_o^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

B_o is the buoyancy number evaluated at the center of the outlet.

For the range of conditions for which Equation 2 applies, B_o may be set equal to $1.04\,\overline{B}_o$. Making this substitution:

$$L_{max}/D_o = 1.70\,\overline{B}_o^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

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[1] Exponent numerals refer to References

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1954.

OBJECT OF TESTS

The present paper gives experimentally determined data on the down throws of jets from a vertical discharge unit heater the construction of which conformed to that of units being manufactured and used.

For throws in excess of approximately 8 orifice diameters, these data were correlated with data presented in the second[2] of the two previous papers by means of the following modification of Equation 3:

$$L_{max}/D_o = 1.70 f \overline{B}_o^{\frac{1}{4}} \quad . . . . . (4)$$

Here f is an arbitrarily introduced empirical throw factor determined by test for each of three differently shaped outlets and each of two differently matched unit-heater fans.

Equations 4*a*, 4*b*, 4*c*, and 4*d*, are useful in solving for the various terms in Equation 4.

$$\overline{B}_o = (\overline{U}_o^2/gD_o)/[(\overline{T}_o - T_a)/T_a] \quad . . . . . (4a)$$

$$\overline{U}_o = Q_o/60\, A_o \quad . . . . . (4b)$$

$$L_{max} = 0.30 f\, \overline{U}_o\, [(D_o T_a)/(\overline{T}_o - T_a)]^{\frac{1}{2}} \quad . . . . . (4c)$$

$$D_o = 0.034\, (f\, Q_o/L_{max})^{\frac{1}{2}}\, [T_a/(\overline{T}_o - T_a)]^{\frac{1}{2}} \quad . . . . . (4d)$$

Three examples in the use of Equations 4 to 4*d* will be given.

Example 1: Given the following data, evaluate $\overline{B}_o$ and f:

L_{max} = throw = 21.8 ft.
Q_o = flow of air at outlet = 1410 cfm.
D_o = diameter of outlet = 1 ft.
A_o = area of outlet = 0.785 sq ft.
$\bar{t}_o$ = average temperature of air at outlet = 200 F.
$\overline{T}_o$ = average absolute temperature of air at outlet = 660 F.
t_a = temperature of room air = 70 F.
T_a = absolute temperature of room air = 530 F.

From Equation 4*b*:

$$\overline{U}_o = 1410/(60 \times 0.785) = 30 \text{ ft per sec.}$$

From Equation 4*a*:

$$\overline{B}_o = [30^2/(32.2 \times 1)]/[(660 - 530)/530] = 114.$$

From 4:

$$f = \frac{L_{max}/D_o}{1.70\, \overline{B}_o^{\frac{1}{4}}} = \frac{21.8/1}{1.70\, \sqrt{114}} = 1.20$$

Example 2: Given the following data, determine diameter of outlet required. Heat to be supplied by unit heater = 158,000 Btu/hr.

Quantity of standard, 70 F air to be supplied by unit = 1130 cfm.

L_{max} = down throw of jet = 21.8 ft.
f = throw factor = 1.20.
Mass rate of flow at outlet = 1130 × 0.075 = 84.7 lb/min.
Temperature elevation of outlet air
= (Heat supplied, Btu/hr)/[60 × (outlet air, lb/min) × (specific heat of air)]
= 158,000/[60 × 84.7 × 0.24] = 130 F.
$\bar{t}_o$ = average temperature of outlet air = 70 + 130 = 200 F.

$\overline{T_o}$ = average absolute temperature of outlet air = 200 + 460 = 660 F absolute.
T_a = absolute temperature of room air = 70 + 460 = 530 F absolute.
Q_o = volume rate of flow at outlet = $(\overline{T_o}/530) \times 1130 = (660/530) \times 1130$ = 1410 cfm.

From Equation 4*d*:

$$D_o = 0.034\ (1.20 \times 1410/21.8)^{\frac{1}{3}}\ [530/(660 - 530)]^{\frac{1}{3}} = 1 \text{ ft.}$$

Example 3: Given all of the data under *Example 2* except the down throw, L_{max}, determine L_{max}.
From Equation 4*b*:

$$\overline{U_o} = 1410/60 \times 0.785 = 30 \text{ ft per sec.}$$

From Equation 4*c*:

$$L_{max} = 0.30 \times 1.20 \times 30\ [1 \times 530/(660 - 530)]^{\frac{1}{3}} = 21.8 \text{ ft.}$$

PERFORMANCE FACTORS

In order to evaluate the effectiveness with which the jets utilized initial down thrust and outlet air horsepower to produce throw, two performance factors, in addition to f, have been introduced. They are:

1. Down-thrust ratio, defined as the ratio of the down thrust of the actual jet to that of a standard jet* (subsequently defined) which produces the same throw in feet as the actual jet produces; and

2. Jet-efficiency ratio, defined as the ratio of the outlet air horsepower of the standard jet to that of the actual jet.

GLOSSARY OF TERMS

Down Thrust = $\int \frac{\rho U^2}{g_c}\, dA + \int (\Delta P)\, dA$

Air Horsepower = $\left[\int \frac{\rho U^3}{2g_c}\, dA + \int (\Delta P)\, U\, dA\right] \Big/ 550$

Relative Down Thrust = ratio of actual down thrust to down thrust that a flat velocity profile would develop with the same flow and the same outlet area.

Relative Air Horsepower = ratio of actual air horsepower to the air horsepower that a flat velocity profile would develop with the same flow and the same outlet area.

Standard Jet = an idealized jet that: (1) has a flat velocity profile at the outlet, (2) develops a value of f equal to unity, (3) delivers the same amount of outlet air and heat as the actual jet to be compared to it delivers, and (4) produces the same throw in feet as the actual jet produces. For the standard jet, α_1, β_2, and β_3^2 are each equal to unity.

Down-Thrust Ratio = Ratio of outlet down thrust of actual jet to outlet down thrust of standard jet = $\beta_2/f^{4/3}$.

Jet-Efficiency Ratio = Ratio of outlet air horsepower of standard jet to outlet air power of actual jet = $f^{3/3}/\beta_3^2$.

Mass-mean temperature = $\int[(\rho\ U\ T)\ dA/M_o]$
= [Heat Input. Btu/hr/ $(0.24 \times 3600 \times M_o)$] + T_a.

Bottom of jet = point at which a thermocouple and a Prandtl tube simultaneously moved upward along axis of jet from floor level gives first distinct evidence of entering the jet.

* See Glossary for meaning of terms marked with an asterisk.

Values of these ratios and of f, together with other pertinent data, are given in Table 1, which presents the principal results of the studies completed on a unit heater operated as described in the following section.

EQUIPMENT AND PROCEDURES EMPLOYED

Two series of tests were run. In the first series, a unit-heater fan was employed which subsequently was found to have excessive tip clearances for

NOMENCLATURE

a = a dimensionless entrainment factor.
A = area, square feet. In Figs. 10, 11, and 12, A is portion of outlet over which corresponding ordinate has been exceeded, square feet.
A_o = area of outlet, square feet.
A_e = effective area of outlet, square feet.
B_o = buoyancy number evaluated at center of outlet, $(U_o^2/gD_o)/[(T_o/T_a) - 1]$.
$\overline{B_o}$ = cross-stream average buoyancy number at outlet, $(\overline{U_o}^2/gD_o)/[(\overline{T_o}/T_a) - 1]$.
D_o = orifice diameter of outlet, feet.
dA = differential area at outlet, square feet.
f = throw factor, used in Equation 4.
g = gravitational acceleration, 32.2 feet per (second) (second).
g_c = 32.2 [(lb − mass/lb − force)] (ft/sec^2).
L_{max} = distance from outlet to bottom (see Glossary) of jet, feet.
M_o = mass rate of flow at outlet, pounds per second.
o = subscript denoting value at outlet.
$\Delta P = P - P_a$ = local static pressure at outlet in pounds per square foot, gage.
P = local pressure at outlet, pounds per square foot, absolute.
P_a = atmospheric pressure at level of outlet, pounds per square foot, absolute.
Q_o = volume rate of flow at outlet, cubic feet per minute.
T = local temperature at outlet, Fahrenheit, absolute.
$\overline{T_o}$ = mass-mean temperature at outlet, Fahrenheit, absolute.
T_a = temperature of undisturbed room air, measured at instrument table approximately 30 in. above floor, Fahrenheit, absolute.
U = local velocity at outlet, feet per second.
U_o = value of U at center of outlet.
$\overline{U_o}$ = cross-stream average velocity at outlet = volume rate of flow/area of outlet, feet per second.
U_{om} = maximum local velocity at outlet, feet per second.
α_1 = ratio of average to maximum outlet velocity = $\overline{U_o}/U_{om} = (\int U dA)/(U_{om}A_o)$.
β_2 = relative outlet down thrust =

$$\left[\int \frac{\rho U^2 d A}{g_c} + \int (\Delta P)\, d A\right] \Big/ \frac{M_o \overline{U_o}}{g_c}$$

β'_2 = relative outlet momentum force = $\int \frac{\rho U^2 d A}{M_o \overline{U_o}}$

$\beta_3{}^2$ = relative outlet air horsepower =

$$\left[\int \frac{\rho U^3 d A}{2g_c} + \int (\Delta P)\, U d A\right] \Big/ \frac{M_o \overline{U_o}^2}{2g_c}$$

$(\beta'_3)^2$ = relative outlet flux of kinetic energy = $\int \frac{\rho U^3 d A}{M_o \overline{U_o}^3}$

ρ = local mass density at outlet, pounds per cubic foot.
$\overline{\rho_o}$ = mean mass density at outlet, pounds per cubic foot.
θ = angle of spread at the outlet.

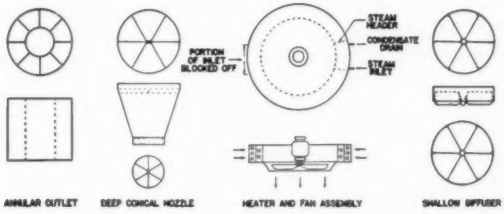


FIG. 1. SCHEMATIC DRAWING OF HEATER AND OUTLET

the outlets employed. In the second series of tests, an appropriate fan was employed. In view of the purpose of the studies conducted, the data from both series of tests are presented.

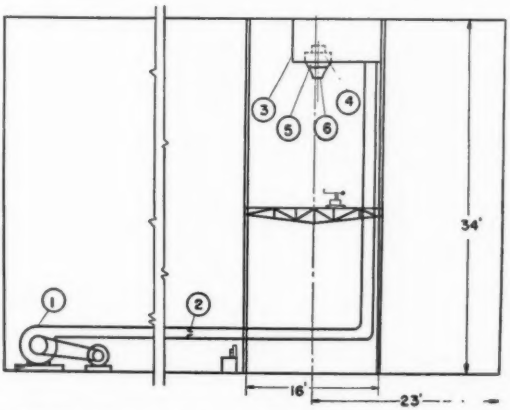
In the first series of tests, two types of outlets were employed:

1. A deep conical nozzle with fixed guide vanes; and
2. A shallow diffuser with adjustable guide vanes which were set vertical.

In the second series of tests, two types of outlets also were employed:

1. The deep conical nozzle employed in the first series of tests; and
2. An annular outlet formed by two concentric cylinders with intervening vertical guide vanes.

The outlets, the heater, and the set-up of equipment are shown in Figs. 1 and 2. The techniques mainly used are those described in the previous papers.[1, 2] In particular, the bottom of the jet was established by simultaneously moving both a thermocouple and a Prandtl-type impact tube upward into the jet from the



Reference numbers: (1) blower, (2) measuring nozzle, (3) plenum chamber, (4) heater, (5) fan, (6) outlet.

FIG. 2. SCHEMATIC ARRANGEMENT OF EQUIPMENT

floor level and noting the point at which both gave evidence of entering the jet. In the second series of tests, impact-tube and static pressure differentials less than 0.01 in. of water were measured and recorded by means of a newly developed, highly sensitive, electronic, pressure-differential recorder, and this recorder was also used to locate the point of maximum downward travel of the jet. Although independently developed at Kansas State College, this instrumentation is similar in basic design to instrumentation developed by the National Bureau of Standards and described in the Bureau's Report 2165.[3] All temperatures for both series of tests were recorded by means of a sensitive, high-speed, electronic recorder.

The unit heater was placed in a plenum chamber to which a measured flow of unheated air was delivered by means of an external blower. The unit-heater

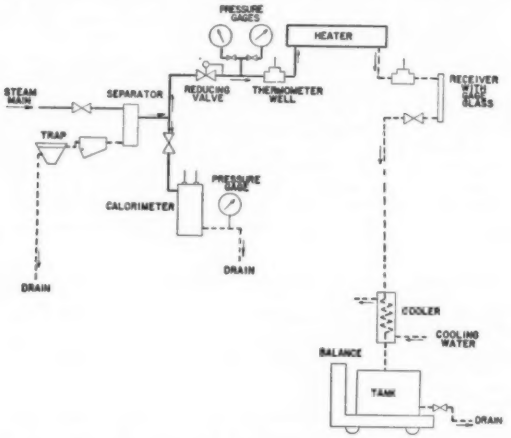


FIG. 3. SCHEMATIC ARRANGEMENT OF EQUIPMENT FOR MEASURING HEAT DELIVERY TO JET

fan was operated at each of three speeds: full speed (1100 *rpm*), three quarters speed (800 *rpm*), and idling, *i.e.*, with the fan power shut off. Under the latter condition, the flow of air through the heater was sustained solely by the external blower. For normal service conditions, atmospheric pressure was maintained in the plenum chamber. In the first series of tests, data on throws beyond the normal range of the heater fan were obtained by employing higher than atmospheric pressure in the plenum chamber.

Air entered the heater radially. In order to obtain an approximately symmetrical jet, it was found necessary to compensate for the segment of the air-inlet area of the heater that was blocked off by the steam header where the header entered the heater. This was accomplished by blocking off a corresponding segment of the air-inlet area directly opposite the header. (See Fig. 1)

Temperatures recorded as outlet temperatures are mass mean temperatures (see Glossary). In the initial sequence of tests, during which the original fan and the deep conical nozzle were employed, the mass-mean outlet temperature

TABLE 1—TEST RESULTS FOR NORMAL SERVICE CONDITIONS AND FOR EQUIVALENT FLOWS WITH FAN IDLING

DATA	SHALLOW DIFFUSER		DEEP CONICAL NOZZLE			ANNULAR OUTLET	
	Fan No. 1 In Service[a]	Fan No. 1 Idling	Fan No. 1 In Service	Fan No. 2 In Service[a]	Fan No. 1 Idling	Fan No. 2 In Service	Fan No. 2 Idling
BASIC							
Orifice diam D_o, ft	1.44	1.44	1.14	1.14	1.14	1.44	1.44
Flow at outlet, cfm	894	903	707	1,070	698	1,580	1,620
Heat Delivery, Btu/hr	49,000	53,000	40,400	50,500	45,000	75,000	78,700
Input, to motor, kw	0.159	———	0.164	0.156	———	0.170	———
Static Press. in plenum chamber, in. water	0	0.088	0	0	0.081	0	0.286
Avg Outlet Temp, F	141	143	140	132	140	121	120
Avg Room Temp, F	82	79	78	80	70	71	70
Avg Outlet Vel, fps	9.15	9.24	11.55	17.40	11.40	16.20	16.60
Avg Outlet Vel, fpm	549	554	693	1,040	684	972	996
Throw, L_{max}, ft	11.5	10.5	13.5	18.0	10.3	20.0	18.5
Angle of Spread, deg[c]	31	26	18	18	18	20	20
Avg Static Press. at Outlet, in. water	−0.022	−0.001	−0.002	+0.001	−0.001	−0.007	−0.004
DIMENSIONLESS OUTLET FACTORS							
Buoyancy Number, B_o	16.6	15.5	31.6	86.1	26.8	60.1	63.3
Fraction of Area Effective, A_e/A_o	0.48	0.91	0.75	0.99	1.00	0.80	0.96
Ratio of Avg Vel to maximum velocity, α_1	0.25	0.67	0.36	0.57	0.93	0.51	0.73
Relative Outlet Downthrust, β_2	1.68	1.14	1.59	1.13	0.98	1.35	1.03
Relative Outlet Momentum Force, β'_2	2.36	1.16	1.63	1.12	1.00	1.42	1.07
Relative Outlet Air Horsepower, $\beta_3{}^2$	5.19	1.40	2.95	1.36	0.97	1.90	1.12
Relative Flux of Kinetic Energy, $(\beta'_3)^2$	6.55	1.44	3.03	1.34	1.02	2.03	1.19
Relative Extent of Whirl[b]	4	1	3	2	1	3	1
Relative Irregularity of Vel Distribution[b]	4	1	3	1	0	2	1
PERFORMANCE							
L_{max}/D_o	8.0	7.2	11.8	15.8	9.1	13.9	12.9
Throw factor, f	1.15	1.08	1.24	1.00	1.03	1.05	0.96
Down-thrust Ratio	1.40	1.03	1.20	1.13	0.94	1.26	1.08
Jet Efficiency Ratio	0.28	0.88	0.60	0.74	1.11	0.60	0.80

[a] Fan No. 1: Original Fan; Fan No. 2: Replacement Fan.
[b] Numbers represent order in which magnitudes fall: 4 represents the largest magnitude observed, 1 the minimum. Zero implies that no whirl or irregularity in the velocity distribution was observed.
[c] Angle of spread of *unheated* jets.

was obtained principally by means of a circular grid made up of approximately 50 thermocouples and calibrated in place against heat-balance data. In all subsequent tests, including all of the tests listed in Table 1, the outlet temperature was evaluated directly from heat-balance data. However, the circular grid of thermocouples was continuously employed to check the temperatures at the

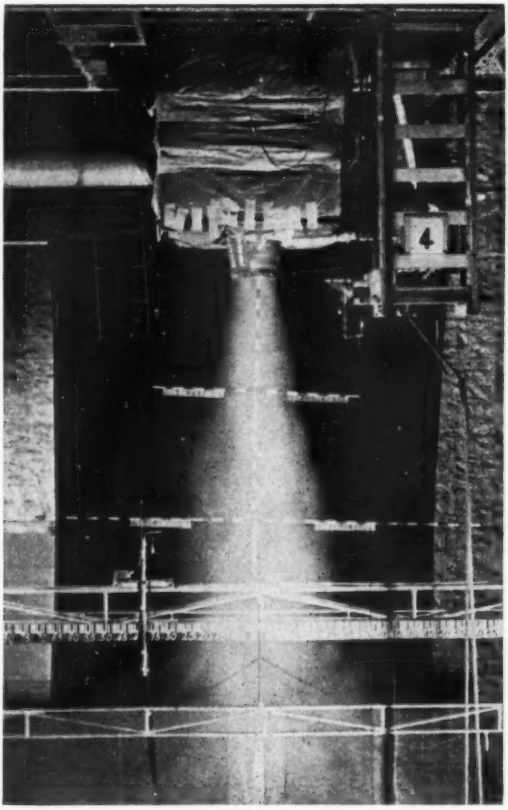


FIG. 4. JET FROM DEEP CONICAL NOZZLE WITH ORIGINAL FAN AT FULL SPEED

outlet. The grid measured directly the cross-stream, time-mean temperatures. These agreed with mass-mean temperatures evaluated from heat-balance data only when the outlet-velocity profile was fairly flat.

Fig. 3 shows schematically the arrangement of equipment employed for measuring the rate of heat delivery to the jet. The enthalpy of the steam entering the heater was determined by means of a throttling calorimeter placed ahead

of the reducing valve. Condensate from the heater was accumulated over a period of 45 min to 1½ hr and then weighed.

The angle of spread of the heated jet could not be measured satisfactorily due principally to conditions produced at the outlet by both entrainment and the recirculation of heated air from the base of the heated jet. The angles of spread

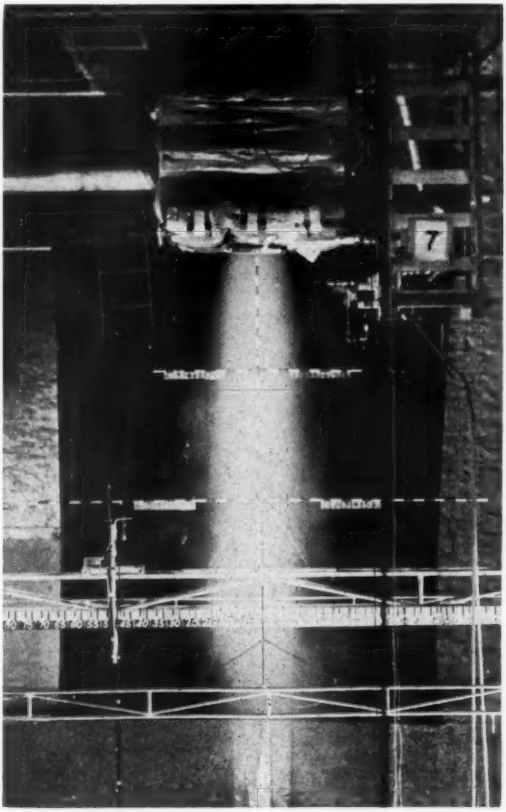


FIG. 5. JET FROM SHALLOW DIFFUSER WITH ORIGINAL FAN AT FULL SPEED

reported for the heated-air jets are, in fact, angles of spread of corresponding isothermal jets. These were approximated visually, using the scale shown in Figs. 4, 5, and 6, and by means of the smoke-gun technique described in the previous paper[2] on jets from *A.S.M.E.* nozzles. These means are recognized as not being entirely reliable, but they were deemed to provide a qualitative indication of the effect of the heater fan and the outlet on the angle of spread.

Prandtl-tube determinations of velocity and static pressure were recognized as being unreliable under the outlet conditions that prevailed when the heater fan was operated. For this reason, Prandtl-tube determinations of outlet velocities were used only to establish the distribution and relative magnitude of the local velocities at the outlet. Mass rates of flow determined from Prandtl-

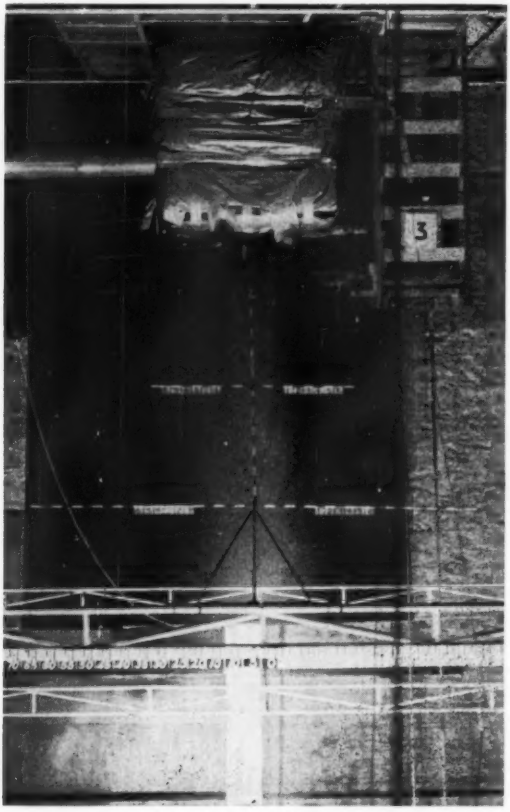


FIG. 6. JET FROM ANNULAR OUTLET WITH REPLACEMENT FAN AT FULL SPEED

tube readings at the outlet differed from the measured mass rate of flow to the plenum chamber by an average of 4 percent and a maximum of 7 percent. In all but one case, that in which the maximum of whirl occurred, static pressure readings were of such a small order of magnitude that they might have been neglected. With one exception, the measured values of static pressure were

negative. Correct static pressures may be regarded as being not less than the measured values.

CHARACTER OF JETS

Figs. 4 to 6 showing the jet made visible by smoke were obtained when the heater fan was operated at full speed. However, no characteristic difference



FIG. 7. JET FROM HEATER WITH NO ATTACHED OUTLET WITH FAN AT FULL SPEED

could be distinguished between these photographs and others taken when the fan was allowed to idle. Fig. 7 shows the type of jet obtained when no attached outlet was employed and the fan was operated at full speed.

Fig. 8 shows velocity and temperature profiles down stream from the outlet, obtained during the first series of tests under the conditions stated in Table 1.

Fig. 9 shows corresponding velocity and temperature profiles obtained during the second series of tests. Where profiles of velocities precisely at the outlet are not shown, these velocities were too irregularly distributed over the area of the outlet to be represented by a single cross-diameter profile.

The annular outlet emitted an annular jet with a stagnant core when the heater fan was in service, and a partly active core when this fan was allowed to idle. This accounts largely for the shapes of the velocity profiles obtained in the vicinity of the annular outlet. In the case of the other outlets, distortions of the velocity profiles in the vicinity of the outlet were due to the heater fan, or to the fan motor and fan hub at the central axis of the outlet. In all cases, velocity

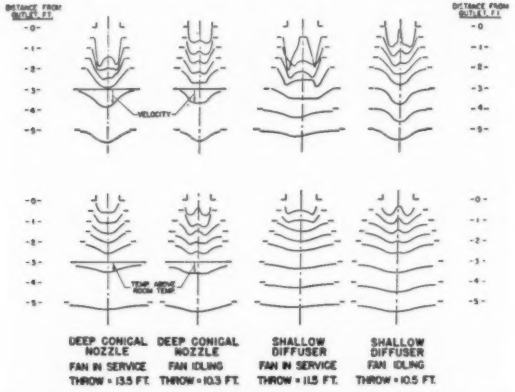


FIG. 8. VELOCITY AND TEMPERATURE PROFILES FOR HEATER EQUIPPED WITH ORIGINAL FAN

profiles underwent progressive change downstream from the outlet and assumed an approximately fully developed form several orifice diameters below the outlet.

OUTLET CONDITIONS

At the outlet one group of factors having a bearing on throw could be evaluated numerically; the other group could not. The first group included:

1. The apex angle or angle of spread of the issuing jet.
2. The fraction of the gross area of the outlet that was dead area, *i.e.*, area unoccupied by down-flowing air.
3. The force or down thrust of the jet as it left the outlet.
4. The air horsepower of the jet as it left the outlet.

The second group, which was made up of factors which could be evaluated only qualitatively, included:

1. The degree of whirl, turbulence, and reversals of the flow in the issuing jet.
2. The degree of irregularity in the distribution of velocities of different magnitudes over the area of the outlet and the amount of mixing that consequently must have occurred to form a fully developed profile.

3. Conditions external to the jet in the vicinity of the outlet produced by the external shape of the outlet and the manner in which it was suspended.

Angles of spread tabulated in Table 1 are angles of spread measured on unheated jets having the same volume rates of flow as the corresponding heated jets. The angles so measured should have relative significance, but they probably are not identical with the angles of spread actually developed by the heated jets. Angles of spread for all outlets other than the short diffuser were between 18 and 20 deg. The angle of spread for the short diffuser was 26 deg with the heater fan idling, 31 deg with the heater fan in service.

The relative extent of whirl and the degree of irregularity in the distribution of velocity, or flow, at the outlet are indicated in Table 1. Whirl was observed

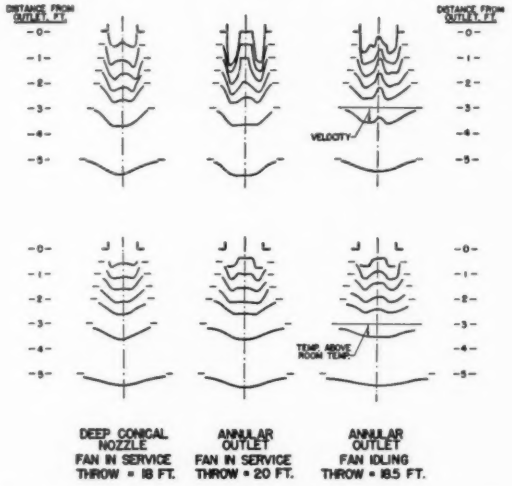


FIG. 9. VELOCITY AND TEMPERATURE PROFILES FOR HEATER EQUIPPED WITH REPLACEMENT FAN

by means of a vertical-bladed pin wheel. For outlets other than the annular outlet, the fraction of dead area at the outlet was felt to be somewhat indicative of the amount of whirl present. Evaluations of the degree of irregularity in the distribution of outlet velocities were based on point to point measurements of velocity at the outlet. No attempt was made to explore conditions external to the jet in the vicinity of the outlet or to evaluate the relative order of turbulence and reversals of flow at the outlet.

To evaluate the down thrust and air horsepower of the jet where it left the outlet, velocities were measured at the centers of 1-in. squares over the accessible area of the orifice which included from 80 to 90 percent of the gross area. These velocities were then plotted in a sequence of decreasing magnitudes on a base of traversed outlet area and replotted to form the curves shown in Figs. 10 to 12. In this procedure, the velocity distribution over the inaccessible area was treated

as being the same as that over the accessible area. Curves corresponding to those of Figs. 10 to 12 were constructed, but not reproduced, for (1) the downward flux of mass per unit area, ρU; (2) the downward flux of momentum per unit area, ρU^2 and (3) the downward flux of kinetic energy per unit area, $(\rho U^3/2)$.

From these curves and data on static pressure at the outlet, the following three factors were obtained (see Nomenclature for defining equations):

α_1 = Ratio of average outlet velocity, $\overline{U}_o$ to maximum local velocity, U_{om}.
β_2 = Relative down thrust of jet = ratio of down thrust at outlet to down thrust that would have occurred had the outlet velocity been uniformly $\overline{U}_o$ and the static pressure uniformly zero.
β_3^2 = Relative air horsepower at outlet = ratio of air horsepower of issuing jet to air horsepower that would have been available had the outlet velocity been uniformly $\overline{U}_o$ and the static pressure uniformly zero.

β_2 is a measure of the force with which the jet issues from the outlet and therefore an index of the capacity of the jet to force itself down against a

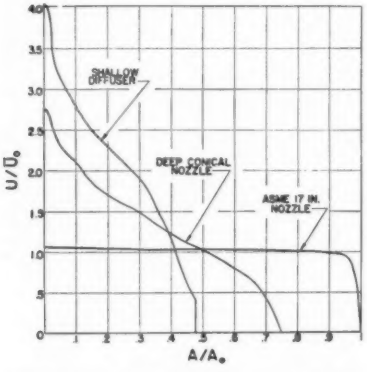


FIG. 10. AREA-DISTRIBUTED VELOCITIES IN SEQUENCE OF DECREASING MAGNITUDES FOR HEATER WITH ORIGINAL FAN

resisting buoyancy force; β_3^2 is an index of the demand for effective fan power at the outlet. For free jets from outlets that produce an approximately flat velocity profile, β_2 and β_3^2 will each be close to unity; for free jets from other outlets, β_2 and β_3^2 will normally be greater than unity. Values of α_1, β_2, and β_3^2 are given in Table 1.

PERFORMANCE CRITERIA

For throws in excess of approximately 8 orifice diameters, two empirical performance criteria were employed:

1. Down-thrust ratio, $\frac{\beta_2}{f^{4/3}}$, and

2. Jet-efficiency ratio, $\frac{f^{8/3}}{\beta_3{}^2}$. The derivations of these criteria are given in the Appendix to this paper.

To establish the foregoing criteria, a standard reference jet was defined. This jet was taken as an idealized jet for which α_1, β_2, $\beta_3{}^2$, and f were each equal to unity. In addition, the standard jet was defined as one that would deliver the same amount of air and the same amount of heat as the actual

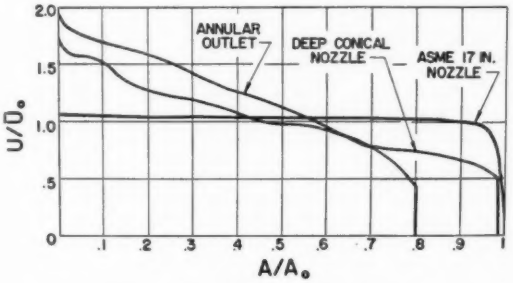


FIG. 11. AREA-DISTRIBUTED VELOCITIES IN SEQUENCE OF DECREASING MAGNITUDES FOR HEATER WITH REPLACEMENT FAN

jet delivered and would produce the same throw in feet as the actual jet produced. Then:

down-thrust ratio = ratio of outlet down thrust of actual jet to outlet down thrust of standard jet.
jet-efficiency ratio = ratio of outlet air horsepower of standard jet to outlet air horsepower of actual jet.

The down-thrust ratio is a measure of the relative magnitude of the force with which the actual jet issues from the outlet to produce a given throw in feet. The jet-efficiency ratio is an index of the relative efficiency with which the actual jet utilizes outlet air horsepower to produce throw. Values of the foregoing criteria are given in Table 1.

TEST RESULTS

Figs. 13 and 14 show plots of f against volume rate of flow based upon data obtained from the series of tests in which the original, or improperly matched fan was employed.

Fig. 13 applies to the heater equipped with the deep conical nozzle. Fig. 14 applies to the heater equipped with the shallow diffuser.

To increase the flow beyond the capacity of the fan, a positive static pressure was established in the plenum chamber and progressively increased. Therefore, as the flow increased, the proportion of the flow generated by the fan decreased.

With the fan in service, the static pressure in the plenum chamber was 0 at 600 cfm for the deep conical nozzle, and was increasingly positive for flows progressively greater than 600 cfm and negative for flows less than 600 cfm.

With the fan in service, the static pressure in the plenum chamber was 0 at 760 cfm for the shallow diffuser, and was increasingly positive for flows progressively greater than 760 cfm.

Positive static pressures were maintained in the plenum chamber when the fan was idling with either the conical nozzle or the shallow diffuser. Throws less than 8 orifice diameters are indicated.

The curves in Figs. 13 and 14 show that:

1. The value of f for the conical outlet decreased in magnitude and approached unity as the volume rate of flow increased; and

2. The values of f for the shallow diffuser were lower in magnitude but similar in trend to those of the conical outlet.

In respect to the first of these effects, the inference may reasonably be drawn that the decrease of f with an increase of flow was associated with a decrease in the relative down thrust, β_2. This inference is supported by data in Table 1.

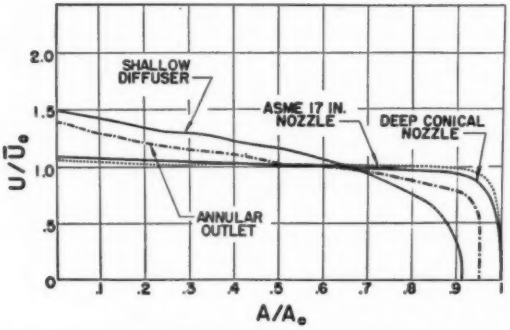


FIG. 12. AREA-DISTRIBUTED VELOCITIES IN SEQUENCE OF DECREASING MAGNITUDE FOR HEATER WITH IDLING FAN

The data in this table show that when the flow was generated solely by the fan, β_2 had a value substantially in excess of unity; when the flow was generated solely by an elevation of plenum-chamber pressure, β_2 had a value that approximated unity. In respect to the second of the two effects the inference may be drawn that the lower values of f developed by the shallow diffuser were associated with the substantially larger angle of spread, θ, developed by this outlet.

Table 1 gives data obtained under normal service conditions, *i.e.*, with atmospheric pressure in the plenum chamber, and comparable data obtained with the heater fan idling and a positive static pressure in the plenum chamber. Values of β_2 and $\beta_3{}^2$, uncorrected for static pressure at the outlet, are tabulated in Table 1 as β'_2 and $(\beta'_3)^2$, along with properly evaluated values of β_2 and $\beta_3{}^2$. These uncorrected values are respectively the relative momentum force at the outlet and the relative flux of kinetic energy at the outlet. The difference between the corrected and uncorrected values is significant only for the case wherein the shallow diffuser was employed with the fan in service. In this case, the correction employed was probably excessive, since flow conditions at the

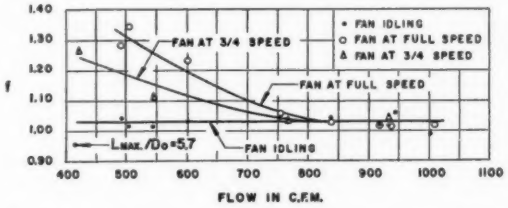


FIG. 13. VALUES OF f DEVELOPED BY THE DEEP CONICAL NOZZLE EQUIPPED WITH ORIGINAL FAN

outlet were conducive to an inaccurately large static pressure reading in a negative direction.

Previous studies on jets from *A.S.M.E.* nozzles, indicated that deviations in f of ± 10 percent should be expected under apparently equivalent conditions. It follows that only two values of f deviated significantly from unity. These values were obtained with the original fan in service and were associated with high values of β_2 and $\beta_3{}^2$.

The conical outlet, equipped with the replacement fan, produced a flow 50 percent greater and a throw in feet 33 percent greater than the flow and throw produced by the same outlet equipped with the original fan. The annular outlet in turn delivered a flow 50 percent greater than the conical outlet delivered, and produced a throw in feet 10 percent greater than the latter produced. The maximum power demand of the fan motor was 0.170 kw, the minimum demand 0.156 kw, a difference of less than 10 percent.

In Table 2, significant data from Table 1 are retabulated to show the effects of converting the fan from idling to service operation.

CONCLUSIONS

1. The performance of a unit heater will depend upon constructional, operational, and installation features. Performance data presented, therefore, should not be used indiscriminately, nor should they be used as being typical of either unit heaters in general or of the particular unit heater tested.

2. The separate effects of basic outlet factors, β_2 $\beta_3{}^2$, θ etc., on the throw factor (f) could not be evaluated quantitatively. To accomplish this, a fundamental type of research must be pursued.

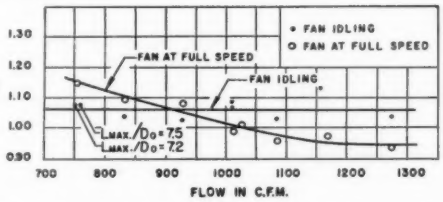


FIG. 14. VALUES OF f DEVELOPED BY THE SHALLOW DIFFUSER EQUIPPED WITH THE ORIGINAL FAN

3. Although the separate effects of basic outlet factors could not be evaluated quantitatively, the data indicate that:

a. An increase in the relative down thrust, β_2, and consequently an increase in the relative outlet air horsepower, $\beta_3{}^2$, promotes an increase in f.

b. A decrease in the effective area of the outlet and an increase in whirl each promotes an increase in the demand for down thrust and air power required at the outlet to maintain a fixed throw in feet. Thus the annular outlet, with 80 percent of the outlet area effective, developed a down-thrust ratio 12 percent greater than that developed by the conical outlet, with 99 percent of its outlet area effective, and a jet-efficiency ratio 19 percent less than the latter developed. The shallow diffuser, with 48 percent of its area effective and the highest level of whirl, developed the highest down-thrust ratio and the lowest jet-efficiency ratio, therefore the highest relative demand for outlet down-thrust and air horsepower.

TABLE 2—EFFECT OF FAN IN SERVICE *vs.* FAN IDLING

TERM	FAN	SHALLOW DIFFUSER	DEEP CONICAL NOZZLE	ANNULAR OUTLET	AVERAGE
Throw Factor f	No. 1 in Service [a]	1.15	1.24	—	1.20
	No. 2 in Service [a]	—	1.00	1.05	1.03
	Idling	1.08	1.03	0.96	1.02
Relative Outlet Down Thrust β_2	No. 1 in Service	1.68	1.59	—	1.64
	No. 2 in Service	—	1.13	1.35	1.24
	Idling	1.14	0.98	1.03	1.05
Relative Outlet Air hp $\beta_3{}^2$	No. 1 in Service	5.19	2.95	—	4.07
	No. 2 in Service	—	1.36	1.90	1.63
	Idling	1.40	0.97	1.12	1.16
Down Thrust Ratio	No. 1 in Service	1.40	1.20	—	1.30
	No. 2 in Service	—	1.13	1.26	1.20
	Idling	1.03	0.94	1.08	1.02
Jet Efficiency Ratio	No. 1 in Service	0.28	0.60	—	0.44
	No. 2 in Service	—	0.74	0.60	0.67
	Idling	0.88	1.11	0.80	0.93

[a] Fan No. 1—Original Fan
Fan No. 2—Replacement Fan

4. The fan had a pronounced effect on outlet conditions and on the performance of the jet based upon outlet conditions. Thus:

a. The average jet-efficiency ratio was 0.44 with the original fan in service, 0.67 with the replacement fan in service, and 0.93 with the heater fan idling. Corresponding values of the downthrust ratio were 1.30, 1.20, and 1.02;

b. When the improperly matched (original) fan was replaced by a properly matched fan, the jet efficiency ratio for the conical outlet was increased from 0.60 to 0.74, the down-thrust ratio was reduced from 1.20 to 1.13, the effective fraction of the outlet area was increased from 0.75 to 0.99, and the factor, f, was reduced from 1.24 to unity.

5. The most advantageous performance based on outlet conditions was provided, for normal service conditions, by the conical outlet equipped with the replacement fan: down-thrust ratio 1.13, jet-efficiency ratio, 0.74. The least advantageous performance for normal service conditions was provided by the shallow diffuser equipped with the original fan: down-thrust ratio 1.40, jet-efficiency ratio 0.28.

6. The shallow diffuser developed the largest angle of spread, an average value of 28 deg for unheated air. The conical and annular outlets developed angles of spread about the same as those characteristic of unheated jets from an *A.S.M.E.* nozzle.

7. The overall performance of the annular outlet equipped with the replacement fan was superior to that of the other outlets. This outlet delivered 50 percent more air and produced 10 percent more throw in feet than the conical outlet likewise equipped with the replacement fan. The superior performance of the annular outlet must have been due to lowered frictional losses, since the conical outlet was actually superior to the annular outlet in its use of outlet down-thrust and outlet air horsepower.

8. From the foregoing, the following supplementary conclusions may be drawn:

a. The unit heater fan should be carefully matched to the outlet.

b. The outlet and fan combination should be designed for overall effectiveness, that is with regard for both the resistance offered by the outlet to flow and the pattern of flow developed at the outlet. (The external shape of the outlet is also a factor to be considered.)

c. The less the whirl, dead area, and irregularity in the distribution of flow at the outlet, the more effective will be the jet's utilization of outlet down-thrust and outlet air horsepower.

d. The throw of a jet may be increased by increasing the velocity of the jet at the outlet and thereby increasing the outlet down thrust and outlet air horsepower. This may be done, for example, by artificially reducing the effective area of the outlet. Whirl, because of its effect on the effective area, may produce the same result.

e. Increasing f in the foregoing manner may entail an increase in power consumption out of proportion to the effect produced on throw. The factor, f, by itself, is therefore not a sufficient criterion of performance. In addition two criteria, the down-thrust ratio and the jet-efficiency ratio, as employed herein, are needed to evaluate the relative effectiveness with which outlet down thrust and outlet air horsepower are utilized.

RECOMMENDATIONS

1. An idealized jet, which may be approximated in the form of a free jet from an *A.S.M.E.* nozzle installed in accordance with *A.S.M.E.* code specifications[4] is recommended as a standard jet with which practical jets may be compared in terms of performance. For evaluating the outlet down-thrust ratio and the outlet jet-efficiency ratio, this jet would deliver the same amount of air and heat in unit time as the practical jet delivers, and would produce the same throw in feet as the practical jet produces. For throws in excess of 8 orifice diameters, the standard jet would have a flat velocity profile and values of β_2, $\beta_3{}^2$, and f each equal to unity.

2. Fundamental research, planned to obtain data applicable to the design and operation of practical outlets should be continued, making use of the guidance produced by the research reported herein. Basic objectives of such research should be to determine, separately, the effects of:

a. Reducing the effective area of an outlet.

b. Modifying the velocity profile.

c. Enlarging the angle of spread.

d. Artificially introducing whirl.

e. Modifying the shape and suspension of the outlet.

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APPENDIX

Evaluations of Down-Thrust Ratio and Jet-Efficiency Ratio, When Throw Exceeds 8 Orifice Diameters

The down thrust of a jet is defined as the total force of the jet as it issues from the outlet. It is made up of the sum of a downward momentum force, $\int (\rho U^2 dA/g_c)$, and a downward force due to static pressure, $\int \triangle P\, dA$.

The air horsepower of the jet is defined as the sum of the downward flux of kinetic energy at the outlet and the power of the jet due to static pressure at the outlet, both expressed in terms of horsepower. Thus:

Air horsepower $= [\int (\rho\, U^3\, dA/2\, g_c) + \int (\Delta P U dA)]/550.$

Making use of the definitions of β_2 and β_3^2:

Down-thrust of jet $= \beta_2(M_o \overline{U}_o)/\, g_c$, and . . . . . (A-1)

$$\text{Air horsepower of jet} = \beta_3^2\, [M_o \overline{U}_o^2/(2g_c)]/550 \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-2)}$$

Let values for the standard jet be indicated by subscript s. Then:

$$\text{Down thrust of standard jet} = \beta_{2s}\, [(M_o \overline{U}_{os})/(g_c)] \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-3)}$$

$$\text{Air horsepower of standard jet} = (\beta_{3s})^2 \left(\frac{M_o\, (\overline{U}_{os})^2}{2g_c} \right) \Big/ 550 \quad . \quad . \quad \text{(A-4)}$$

From Equations A-1, A-2, A-3 and A-4 and definitions of down-thrust and jet-efficiency ratios:*

$$\text{Down-thrust ratio} = \frac{\beta_2\, (M_o\, \overline{U}_o/g_c)}{\beta_{2s}\, (M_o\, \overline{U}_{os}/g_c)} = \frac{\beta_2\, \overline{U}_o}{\beta_{2s}\, \overline{U}_{os}} \quad . \quad . \quad . \quad . \quad . \quad \text{(A-5)}$$

$$\text{Jet-efficiency ratio} = \frac{(\beta_{3s})^2\, [M_o\, (\overline{U}_{os})^2/2g_c]}{\beta_3^2\, [M_o\, (\overline{U}_o{}^2)/2g_c]} = \left[\frac{\beta_{3s}\, \overline{U}_{os}}{\beta_3\, \overline{U}_o} \right]^2 \quad . \quad . \quad . \quad \text{(A-6)}$$

From Equation 4 of the paper

$$\frac{L_{max}}{D_o} = 1.70 f \overline{B}_o{}^{\frac{1}{2}} = 1.70 f \left[\frac{\overline{U}_o{}^2/g\, D_o}{\left(\frac{\overline{T}_o}{T_a} - 1 \right)} \right]^{\frac{1}{2}}$$

and

$$L_{max} = 0.30 f\, \overline{U}_o \left[\frac{D_o}{\left(\frac{\overline{T}_o - T_a}{T_a} \right)} \right]^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-7)}$$

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* For definitions see Glossary.

For the standard jet, recognizing that L_{max}, $(\overline{T}_o - T_a)$, and T_a are the same for both jets,

$$L_{max} = 0.30\,\overline{U}_{os} f_s \left[\frac{D_{os}}{\left(\frac{\overline{T}_o - T_a}{T_a} \right)} \right]^{\frac{1}{2}} \quad \text{. . . . . (A-8)}$$

From Equations A-7 and A-8

$$\overline{U}_{os} = \frac{f}{f_s} \overline{U}_o \left(\frac{D_o}{D_{os}} \right)^{\frac{1}{2}} \quad \text{. . . . . (A-9)}$$

$$M_o = \text{Mass rate of flow} = \overline{U}_o \bar{\rho}_o \left(\frac{\pi D_o^2}{4} \right) = \overline{U}_{os} \bar{\rho}_{os} \left[\frac{\pi (D_{os})^2}{4} \right] \quad \text{. . . (A-10)}$$

From Equation A-10, since $\bar{\rho}_o = \bar{\rho}_{os}$;

$$(D_o/D_{os}) = (\overline{U}_{os}/\overline{U}_o)^{\frac{1}{2}} \quad \text{. . . . . (A-11)}$$

From Equations A-9 and A-10

$$\overline{U}_{os} = (f/f_s)^{4/3}\, \overline{U}_o \quad \text{. . . . . (A-12)}$$

From Equations A-5 and A-12:

$$\text{Down-thrust ratio} = (\beta_2/\beta_{2s})\, (f_s/f)^{4/3} \quad \text{. . . . . (A-13)}$$

From A-6 and A-12

$$\text{Jet-efficiency ratio} = (f/f_s)^{8/3}\, (\beta_{3s}/\beta_3)^2 \quad \text{. . . . . (A-14)}$$

For standard jets with throws in excess of 8 orifice diameters, β_{2s}, β_{3s}^2, and f_s will be unity. Therefore, for throws in excess of 8 orifice diameters:

$$\text{Down-thrust ratio} = \frac{\beta_2}{f^{4/3}} \quad \text{. . . . . (A-15)}$$

$$\text{Jet-efficiency ratio} = \frac{f^{8/3}}{\beta_3^2} \quad \text{. . . . . (A-16)}$$

Significant characteristics of a standard jet and actual jet may be compared as follows: From Equations A-11 and A-12, by making f_s equal to unity:

$$D_{os}/D_o = 1/f^{\frac{1}{2}} \quad \text{. . . . . (A-17)}$$

$$(L_{max}/D_{os})/(L_{max}/D_o) = D_o/D_{os} = f^{\frac{1}{2}} \quad \text{. . . . . (A-18)}$$

$$\frac{\overline{B}_{os}}{\overline{B}_o} = \frac{(\overline{U}_{os})^2/D_{os}}{U_o^2/\overline{D}_o} = \left(\frac{\overline{U}_{os}}{\overline{U}_o} \right)^{\frac{1}{2}} = f^{10/3} \quad \text{. . . . . (A-19)}$$

A comparison between an actual jet and standard jet may be made as follows:

Let the following data apply to an actual jet:

$f = 1.20$; $\overline{U}_o = 30$ ft/sec; $D_o = 1$ ft; $\overline{T}_o = (200 + 460) = 660$ F absolute.

$T_a = (70 + 460) = 530$ F absolute; $\beta_2 = 1.30$; and $\beta_3^2 = 1.80$. Then:

(*A*) Computations for actual jet:

$$\overline{B}_o = \frac{\overline{U}_o^2/(g \times D_o)}{\left(\frac{\overline{T}_o}{T_a} - 1\right)} = \frac{(30)^2/(32.2 \times 1)}{\left(\frac{660}{530} - 1\right)} = 114$$

$$L_{max}/D_o = 1.70 f \overline{B}_o^{\frac{1}{3}} = 1.70 \times 1.20 (114)^{\frac{1}{3}} = 21.8.$$

$$L_{max} = 21.8 \times 1 = 21.8 \text{ ft.}$$

$$\rho_o = 14.7 \times 144/[53.3 \times (200 + 460)] = 0.060 \text{ lb/cu ft.}$$

$$M_o = \rho_o \overline{U}_o \pi D_o^2/4 = 0.060 \times 30 \times \pi \times 1/4 = 1.41 \text{ lb/sec.}$$

$$\text{Down thrust} = \beta_2(M_o\overline{U}_o)/g_c = 1.30 \times 1.41 \times 30/32.2 = 1.71 \text{ lb.}$$

$$\text{Air horsepower} = \beta_3^2 (M_o\overline{U}_o^2/2\, g_c)/550 = 1.80 \times 1.41 \times (30)^2/(64.4 \times 550) = 0.0645 \text{ hp.}$$

(*B*) Computations for the standard jet:

f_s, β_{2s}, and β_{3s}^2 will each be unity

$$\overline{U}_{os} = f^{4/3} \times \overline{U}_o = (1.20)^{4/3} \times 30 = 38.3 \text{ ft/sec.}$$

$$D_{os} = D_o/f^{\frac{1}{2}} = 1/1.20^{\frac{1}{2}} = 0.885 \text{ ft.}$$

$$\overline{B}_{os} = \overline{B}_o f^{10/3} = 114 \times 1.20^{10/3} = 209.$$

$$L_{max}/D_{os} = L_{max}/D_o \times f^{\frac{1}{2}} = 21.8 \times (1.20)^{\frac{1}{2}} = 24.6.$$

$$L_{max} = 24.6 \times D_{os} = 24.6 \times 0.885 = 21.8 \text{ ft.}$$

$$\text{Down-thrust} = \beta_{2s} M_o \overline{U}_{os}/g_c = 1 \times 1.41 \times 38.3/32.2 = 1.68 \text{ lb.}$$

$$\text{Air horsepower} = [(\beta_{3s})^2 M_o (\overline{U}_{os})^2/2g_c]/550 = 1 \times 1.41 \times (38.3)^2/64.4 \times 550 = 0.0584 \text{ hp.}$$

(*C*) For comparison of actual and standard jets:

$$\text{Down-thrust ratio} = \frac{\beta_2}{f^{4/3}} = 1.02.$$

$$\text{Jet-efficiency ratio} = \frac{f^{3/3}}{\beta_3^2} = 0.90$$

These ratios may also be obtained from preceding computations for actual and standard jets:

Down-thrust ratio = 1.71/1.68 = 1.02

Jet-efficiency ratio = 0.0584/0.0645 = 0.90

DISCUSSION

W. O. HUEBNER, New York, N. Y. (WRITTEN): Vertical discharge unit heaters are used in thousands of industrial plants throughout the country, but we do not know enough about their performance to prevent occasional unsatisfactory application. It is, therefore, necessary to learn more about the basic problems involved, and Professor Helander's investigation serves this end.

This paper is the first to deal quantitatively with the behavior of jets from vertical unit heaters, and it is, therefore, of great practical value. To properly evaluate the paper, it is interesting to compare this with a paper on unit heating presented before the Society 25 years ago. This earlier paper, which discussed the problem entirely in terms of smoke photographs and trained arrows, gave no quantitative information at all, but it was considered a good paper at the time.

We need more exact information today. Professor Helander's throw factor, as well as the down-thrust ratio and the jet-efficiency ratio are useful in dealing with the complex matter of unit heater air distribution.

We should realize that studies of this kind call for time and perseverance, in addition to the necessary scientific qualifications of the investigators. We are fortunate in having the fine cooperation of various institutions to do this kind of research. It is gratifying to know that Professor Helander's air distribution research on unit heaters will be continued. Has the shallow diffuser been used in the test runs with fan No. 2?

G. L. TUVE, Cleveland, Ohio: In this paper Fig. 1 is a schematic drawing showing a number of straightening vanes. Is this an important device that should be a part of the unit heater installation? How did the authors learn that they were necessary?

I. S. DANE, Boston, Mass.: This paper very well describes the downward jet, but as unit heaters have to recirculate air from the floor back up to the unit, I would like to know the characteristics of the action of the air rising to the unit.

W. A. GRANT, Syracuse, N. Y.: I would like to compliment the authors on an extremely fine paper. For the same type of outlet, would the authors expect a blowthrough unit heater to give better performance than a drawthrough unit heater in terms of throw and air horsepower efficiency? I would certainly expect, qualitatively at least, that the blowthrough unit would have a better velocity distribution. The heat transfer performance of a blowthrough on the first row is generally better because of all the turbulence; therefore the overall performance might be improved. I would expect the air horsepower performance to be somewhat better.

W. M. MYLER, JR., Columbus, Ohio: This paper deals with the type of unit heater which is almost universally restricted to steam heating. Is there any contemplation of doing any work with gas-fired unit heaters? They usually employ horizontal discharge and have deflecting vanes to direct the heated air stream to the useful area.

L. R. PHILLIPS, Hartford, Conn.: I wonder how much influence the swirl component from the fan had on the downthrow? Have the authors tried different fans or made any notation on the actual discharge pattern of the fan without the diffuser, making note of velocities and swirl components?

R. C. EDWARDS, Pompton Plains, N. J.: If you are interested in fundamental studies, would it be desirable to take the area normally served by the unit heater and determine the horsepower required to obtain a deviation in the temperature there? A minimum deviation in room air temperature is an important consideration. To simply blow the air down from a unit heater and have it return to the ceiling again is not a good practice. The total horsepower required to run the heater may be less but the discomfort of the heated area results.

If the room were half the size described, and the specifications required that the temperature did not deviate from one side of the room to the other more than 2 deg, what is the minimum horsepower to obtain this minimum deviation?

H. H. YOUSOUFIAN, New York, N. Y.: The authors have presented an excellent paper in which they indicate the throw obtainable from various forms of basic nozzles discharging heated air downward. The designer of a unit heater or a projection heating system is also interested in knowing how the throw from the various nozzles influences the temperature equalization within the zone of occupancy. In other words, a nozzle which is otherwise efficient in producing a long throw may or may not produce good temperature equalization within the zone of occupancy and is therefore

an important consideration in the selection of a nozzle type. I would be interested in knowing what observations, if any, the authors may have made in these respects.

H. E. ZIEL, Detroit, Mich.: Could the formulae be used to determine conditions near the floor line if cooler than surrounding air is supplied through the jet to obtain cooling near the floor line? For example, if a blower fan unit were located in a roof truss space 40 ft above the floor line, the truss space temperature was 105 F and the fan unit would draw air in at 95 F from above the roof, what cooling effect could be expected at the floor line? Would not the whirl be desirable?

What should be done with jets to produce as near as possible a cooling effect on men working at the floorline? Might this be compared with the ability to get heat down to the floorline in the wintertime?

EDWARD BERLY,* Waltham, Mass.: At the mouth of the orifice does the author favor a flat velocity distribution or a peaked velocity?

AUTHORS' CLOSURE (Linn Helander): Some of the questions raised can be answered only conditionally, if at all. However, the questions raised emphasize the need of the research we are doing and will be helpful in determining the direction of future research.

We are particularly pleased to know that the results of our research will be useful to industry and that Mr. Huebner regards with favor the three performance factors we have defined and employed in our paper, namely, the throw factor, the down-thrust ratio, and the jet-efficiency ratio.

Mr. Huebner has asked whether or not the shallow diffuser was used with Fan No. 2. It was not so used. The time available after we received Fan No. 2 did not permit us to employ the shallow diffuser with this fan.

Mr. Grant suggests certain advantages that a blow-through type of unit heater may have over a draw-through type. We have not tested the blow-through type and so we cannot say that the overall performance of the one would be better than the other. We would expect that the outlet velocity profile would be more favorable in the case of the blow-through type because the heater would serve the same purpose as straightening vanes.

Mr. Yousoufian points out that a nozzle designed to produce a long throw may not provide good temperature equalization in the zone of occupancy. In a given case, the throw for which an outlet will be designed will be determined by the elevation of the outlet above the zone of occupancy. An outlet designed to produce this required throw with a minimum of outlet air horsepower may not provide a satisfactory distribution of air over the zone of occupancy or a satisfactory equalization of temperature. At least two factors other than the efficient use of energy to produce throw are involved here. (1) the spread of the jet and (2) the mixing or entrainment capacity of the jet. If either or both are increased, the outlet air horsepower requirements will probably be increased. There is obviously need for investigation, not yet undertaken, of both the manner in which a jet distributes temperature and velocity in the zone of occupancy and the factors that may be controlled to improve this distribution. Terminal temperatures of the jets that we studied were within a few degrees of room temperature, but we did not employ the jets to heat the room into which they were projected.

Mr. Edward's question presumably implies that more power may be required to maintain uniformity of temperature in the zone of occupancy than is required solely for the purpose of blowing the air down to a certain level, *e.g.*, close to floor level. Possibly this is so, but we have no data on this. Certainly the problem of effecting a

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* Research Division, Raytheon Mfg. Co.

satisfactory distribution of temperature with a minimum of power demand is one deserving of study.

Mr. Dane would like to know the action of the air rising to the unit. Knowledge of this action is essential to a complete understanding of the jet and the influences of the jet on the zone of occupancy. We plan to explore the space outside the jet, and will determine the pattern of upward flow. All we can say at present is that there exists a current of upward air that probably affects the behavior of the jet in a significant manner.

Professor Tuve's question concerns the value of the straightening vanes employed in the outlets tested. These vanes came with the outlets and were vertical. We did not put them in and did not determine their value.

Mr. Phillips asks what influence the whirl component at the outlet may have on the down throw. The effect of whirl on throw is somewhat complicated, because whirl may produce dead area at the outlet, *i.e.*, it may reduce the effective area of the outlet. If it does so, the net result may be an augmentation of the downward momentum flux sufficient to develop a throw greater than would be realized with the same volume rate of flow in the absence of whirl, but the cost in terms of energy consumption will be excessive. Although the effect of whirl cannot be evaluated as an independent factor from our test data, the following data taken from Table 1 are noteworthy:

MAGNITUDE OF WHIRL ON SCALE OF 0-4	4	3	2	1
Throw factor f	1.15	1.15	1.00	1.02
Jet-efficiency ratio	0.28	0.60	0.74	0.90
Number of items averaged	1	2	1	3

A low jet-efficiency ratio means a relatively high energy consumption and vice-versa. It would seem that whirl should be avoided so far as practicable.

Mr. Berly's question is whether a flat velocity profile or a peaked velocity profile is preferred? This is a matter we plan to investigate experimentally.

Data from Table 1 for the fan idling are indicative, and are presented here.

TYPE OF OUTLET	ANNULAR	CONICAL
Average Velocity / Maximum Local Velocity	0.73	0.93
Jet-Efficiency Ratio	0.80	1.11

The velocity profile of the conical outlet, as indicated by the ratio of the average velocity to the maximum velocity, was close to being flat; that of the annular outlet deviated significantly from flatness. Other factors than the shape of the velocity profile are involved, and one may question the accuracy of the jet-efficiency ratio of the conical outlet, but the data appear to give preference to a flat velocity profile where economy in the utilization of outlet air horsepower for throw is a primary consideration. A pattern of flow at the outlet that minimizes whirl, eddies, voids, and sharp discontinuities in the velocity profile will probably be a good pattern, regardless of whether the profile is flat or curved. A flat velocity profile will deliver a given rate of flow of air from an outlet of given diameter with a minimum demand for outlet air horsepower. A curved velocity profile will probably develop a larger throw for the same flow, outlet diameter, and heat delivery, but in doing so, it will augment the demand for outlet air horsepower. The same result may be accomplished by

means of a core that blocks off the central part of the outlet, but for the best utilization of outlet air horsepower, this core should be shaped to minimize voids and eddies.

We are planning to study chilled jets projected vertically downward, but so far we have studied only heated and isothermal jets. Until experimental data are available on chilled jets, we suggest that Mr. Ziel employ formulas presented in Professor Koestel's paper*, for determining the approximate floor-level temperature of downward jets of chilled air. We do not see how whirl created at the outlet would be helpful at the floor. The whirl so created disappears a short distance from the outlet.

We are not now planning research of the kind referred to by Mr. Myler, *i.e.*, research on horizontal-discharge, gas-fired unit heaters. Perhaps, after we have completed our studies of vertical jets, some work could be done on horizontal jets of the gas-fired type.

The discussions have clearly indicated that what is needed is a means of predicting the performance of a jet in terms of the service it renders, *i.e.*, in terms of the economy and effectiveness with which it distributes temperature and ventilating air over the space to be served. With the continuance of the fine support and encouragement we are receiving from the Society and the *Industrial Unit Heater Association*, we shall continue our studies with this requirement in mind.

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* Computing Temperatures and Velocities of Vertical Jets of Hot or Cold Air, by Alfred Koestel (A.S.H.V.E. TRANSACTIONS, Vol. 60, 1954, p. 385).

1512

No. 1512

COMPUTING TEMPERATURES AND VELOCITIES IN VERTICAL JETS OF HOT OR COLD AIR

By ALFRED KOESTEL*, CLEVELAND, OHIO

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the Case Institute of Technology, Cleveland, Ohio.

KNOWLEDGE OF THE behavior of free air jets is at present sufficiently developed so that an attempt can be made to formulate equations which describe the performance of non-isothermal jets. Jet performance from the engineering viewpoint can be defined by maximum centerline velocity variation with distance, together with cross-sectional patterns of velocity and temperature. For non-isothermal jets the temperature-difference variation between the jet and the ambient air is of importance.

The overall treatment of the non-isothermal jet is macroscopic in nature in that the internal mechanisms of fluid flow and heat transfer are not considered. The jet performance is analyzed in terms of principles involving conservation of thermal energy and the equating of the buoyant forces to the change in jet momentum. The differences in rates of momentum transfer and heat diffusion are taken into account by the use of an overall effective turbulent Prandtl number.

Although the values of some of the constants of integration cannot be accurately substantiated, sufficient experimental data are on hand so that approximate values can be established. It is hoped that the equations presented here will contribute to the guidance of any future laboratory investigations of non-isothermal jet performance. The detailed mathematical analyses appears in the Appendix.

NON-ISOTHERMAL FREE JETS

Flow of non-isothermal air from a nozzle into a free space will have the following notable features:

1. A velocity profile and a temperature-difference profile which have shapes that can be approximated by an error function type curve (normal probability curve) (see Fig. 1). The temperature-difference profile is flatter than the velocity profile because heat diffuses more rapidly than momentum.

2. A zone exists near the outlet in which the temperature-difference and the velocity measured along the nozzle axis are approximately constant and equal to those existing

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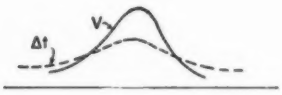


FIG. 1. CROSS-SECTIONAL PROFILE OF VELOCITY AND TEMPERATURE DIFFERENCE FOR A NON-ISOTHERMAL JET

at the outlet (see Fig. 2). This zone is in turn followed by another in which transition takes place from the flow characteristics of a confined fluid to that of a free jet.

3. The free-jet zone, in which the jet is fully developed, is the major zone of the jet.

METHOD OF ANALYSIS

Under isothermal conditions it has been found that the conservation of momentum principle can be applied in the analysis of a free jet since no external forces are assumed to act on the free jet stream. Equations of the form

$$V_c/V_o = \frac{K}{(X_o/D_o)} \quad \ldots \ldots \ldots \ldots \ldots \quad (1)$$

and

$$V_c/V_o = \frac{K + A}{(X_o/D_o) + A} = \frac{K_p}{X/D_o} \quad \ldots \ldots \ldots \quad (2)$$

have been determined from test (see Fig. 3). A is the distance in diameters from the outlet face to an imaginary point source behind the outlet as shown in Fig. 3. Values from 0 to 3 have been reported for A. Equations 1 and 2 are identical when A is zero and K_p becomes equal to K. A can be determined experimentally either by plotting the reciprocal of V_c *vs.* X for the major free-jet zone and extrapolating the curve to zero, or by extension of the angle of expansion to its intersection with the centerline of the nozzle. Equations 1 and 2 apply to the jet flow at distances outside of the constant velocity core. When V_c/V_o equals unity in Equations 1 and 2, $K = (X/D_o)$ and $K_p = (X/D_o) + A$. The constant K, therefore represents the apparent length of the constant velocity zone measured from the nozzle (as determined by the point of intersection Q,

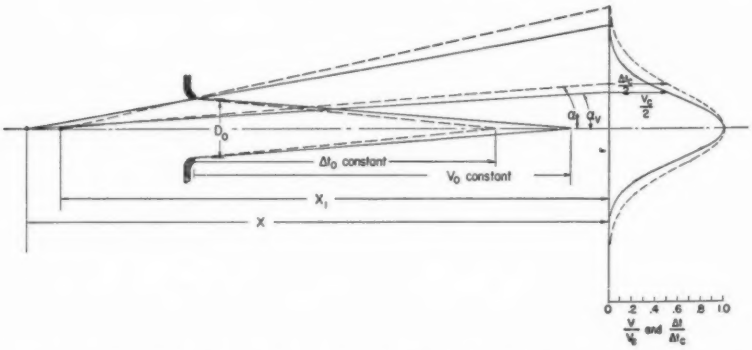


FIG. 2. DIAGRAM OF A NON-ISOTHERMAL JET NEAR THE OUTLET, SHOWING ZONES OF CONSTANT VELOCITY AND CONSTANT TEMPERATURE

Fig. 3), and K_p is the length of the constant velocity zone measured from the point source. K_p is therefore equal to $K+A$. The constants K_p and K have been verified experimentally by Tuve[1] and others and have been found to vary somewhat with outlet velocity, the percentage free area of the outlet (if covered with a grid type lattice), the type of approach to the outlet, and the amount of turbulence induced behind the outlet. For outlet velocities above 1000 fpm a value for K of 6.5 is representative of many tests on nozzles. Differences between the jet density and the density of the receiving medium also affect the core length as shown in Fig. 4.

NOMENCLATURE

A = distance from outlet face to the apparent point source, in diameters. See Fig. 3.

a = shape factor for the velocity profile.

b = shape factor for the temperature difference profile.

C_2 = constant of integration.

D_o = outlet diameter, feet.

g = acceleration due to gravity, feet per (second) (second).

K = distance from nozzle to the termination of the apparent constant velocity core, in diameters. K is the familiar constant of proportionality for air jets[1].

K_p = distance from apparent point source to the termination of the apparent constant velocity core for an isothermal jet, in diameters. ($K_p = K + A$).

$K_{p,n}$ = distance from apparent point source to the termination of the apparent constant velocity core for a non-isothermal jet, in diameters.

N_{Prt} = overall effective turbulent Prandtl number of the non-isothermal jet.

Q_x = heat flow of the jet at distance X measured above ambient temperature, Btu per second.

r = radial distance from centerline axis, feet.

R = $K_{p,n}/K_p$ (see Fig. 4).

Δt_c = maximum centerline temperature difference at distance X, Fahrenheit degrees.

Δt_o = temperature difference at the outlet, Fahrenheit degrees.

$(\Delta t_c/\Delta t_o)_{K_p}$ = temperature difference ratio evaluated at the termination of the apparent constant velocity core at K_p.

V_c = maximum centerline velocity at distance X, feet per second.

V_o = outlet velocity, feet per second.

V = velocity at r, feet per second.

X = distance from the apparent point source based on the velocity profile, feet.

X_1 = distance from the apparent point source based on the temperature difference profile, feet.

X_o = distance measured from the outlet face, feet.

X_{max} = distance to where V_c is equal to zero measured from the apparent point source, feet.

$X_{o(max)}$ = distance to where V_c is equal to zero measured from the outlet face, feet.

α_t = angle of expansion for the temperature difference profile, degrees.

α_v = angle of expansion for the velocity profile, degrees.

β = coefficient of expansion. For a perfect gas $\beta = 1/T_a$, where T_a is the temperature of the ambient air, Fahrenheit absolute.

[1] Exponent numerals refer to References.

SIMPLIFYING ASSUMPTIONS

For vertically projected non-isothermal free air jets an equation similar to Equation 2 can be formulated if the following assumptions are made:

1. The absolute value of the air density is constant throughout the jet, but the differences in density are taken into account when the buoyant forces are considered. This assumption is valid if the temperature differences between the jet and the surroundings are not too great. Temperature differences normally involved in the application of jets to space heating or cooling can be considered not too great.

2. The velocity and temperature difference profiles in the fully developed region of the jet can be represented by an error function type curve or normal probability curve.

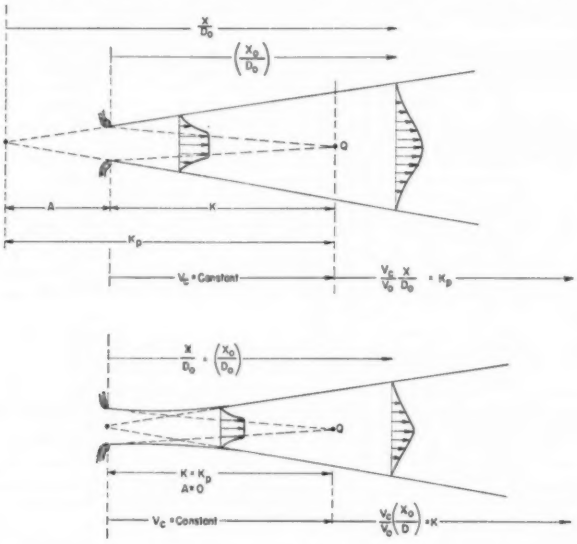


FIG. 3. CHARACTERISTICS OF AN ISOTHERMAL JET AS DEFINED BY EQUATIONS 1 AND 2

3. The difference between the temperature-difference profiles and velocity profiles at a given section of the jet is due to the difference between heat and momentum diffusion in a turbulent jet. This difference in diffusion rate can be expressed by the use of an overall effective turbulent Prandtl number.

4. The angles of expansion for the temperature difference and velocity profiles are constant but differ in value due to the difference in heat and momentum diffusion, or in other words, due to the effective turbulent Prandtl number of the non-isothermal jet.

5. The velocity and temperature difference profiles remain symmetrical about a centerline through the outlet.

6. Thermal energy, Q_X (or enthalpy flow), is conserved in the axial direction. For heated air projected downward and chilled air projected upward a point in the jet may be reached where the jet momentum in the axial direction is reduced to zero by the buoyant forces; and as a result the heat flows radially outward from the jet (see Fig. 5). In this region the assumption that Q_X is a constant and that the velocity and temperature profiles remain symmetrical is not valid.

BASIC EQUATIONS

From the analysis based on the simplifying assumptions already stated, the following expression for the centerline velocity variation with distance from the apparent point source is derived as the equation:

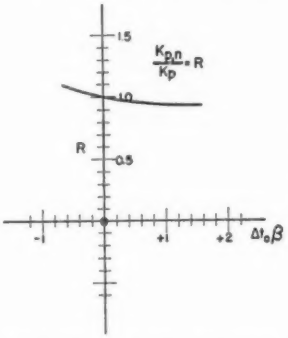
$$(V_c/V_o)^2 = \left(\frac{K_p}{X/D_o}\right)^2 \pm \tfrac{3}{8}C_2\left(\frac{a}{b}+1\right)\left(\frac{D_o\Delta t_o\beta g}{V_o^2}\right)\frac{1}{X/D_o} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The ratio a/b is a function of the overall effective turbulent Prandtl number of the jet as defined by the equation:

$$b/a = \{4/[1+(1/N_{Prt})]\} - 1$$

A Prandtl number of 0.7 has been suggested for non-isothermal jets by such experimenters as Nottage[3], Forstall and Shapiro[4], and Corrsin and Uberoi[5].

For heated jets projected downward and for chilled jets projected upward, the sign is minus in Equation 3; and for heated jets projected upward and chilled jets projected downward the sign is plus. Equation 3 is also applicable



(Computed from data reported by Eck[2])

FIG. 4. EFFECT OF AIR DENSITY ON THE LENGTH OF THE CONSTANT-VELOCITY CORE, MEASURED FROM THE POINT SOURCE

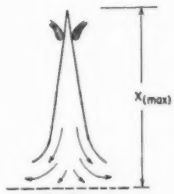


FIG. 5. RADIAL OUT-FLOW FROM A NON-ISOTHERMAL JET

for isothermal jets, for when Δt_o is zero Equation 2 obtains. The dimensionless parameter $D_o\Delta t_o\beta g/V_o^2$ represents the ratio of the buoyant force to the momentum force evaluated at the outlet.

Available test data show a value of 6.5 to 9.5 for K_p, *the apparent* length of the constant-velocity core as determined by the intersection Q, Fig. 3. By considering the conditions at this point Q, *i.e.*, the apparent termination of the constant velocity core, the constant of integration C_2 can be evaluated. For isothermal jets C_2 is theoretically equal to $2K^2$. This can be shown as follows:

For isothermal jets the following equation was derived,

$$V_c/V_o = \sqrt{\frac{C_2}{2}}\frac{D_o}{X} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

when

$V_c/V_o = 1$, $X/D_o = K_p$ and Equation 4 becomes,

$$1 = \sqrt{\frac{C_2}{2}}\frac{1}{K_p} \textit{ and } C_2 = 2K_p^2.$$

From the analysis based on the simplifying assumptions the following equation involving the centerline temperature difference, centerline velocity, and distance from the apparent point source was derived,

$$\frac{C_2(1 + b/a)}{4} = \left(\frac{\Delta t_c}{\Delta t_o}\right)\left(\frac{V_c}{V_o}\right)\left(\frac{X}{D_o}\right)^2 \quad \ldots\ldots\ldots\ldots (5)$$

This equation states that the product of the dimensionless factors defining the centerline temperature difference, centerline velocity and the distance from the point source squared is a constant regardless of the trajectory of the jet. Equation 5 can be used for arriving at a value for C_2 by substituting test data for any non-isothermal jet.

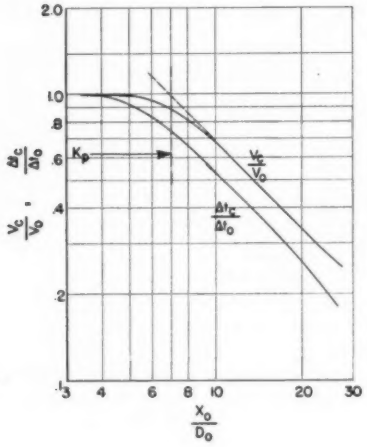


FIG. 6. CENTERLINE VELOCITY AND TEMPERATURE DIFFERENCE AS A FUNCTION OF DISTANCE FROM THE OUTLET FOR A HORIZONTAL JET

(From data reported by Corrsin and Uberoi[5])

When $X/D_o = K_p$ in Equation 5, $V_c/V_o = 1$ and for N_{Prt} of 0.7,

$$C_2 = 2.43\ (\Delta t_c/\Delta t_o)_{Kp}\ K_p^2.$$

From Corrsin and Uberoi data shown in Fig. 6 $(\Delta t_c/\Delta t_o)_{Kp}$ is 0.75 when $X/D_o = K_p$. Therefore, $C_2 = 2.43 \times 0.75\ K_p^2 = 1.82\ K_p^2$.

Data obtained by Nottage on a chilled jet with a $\Delta t_o = 39$ F deg and a $V_o = 500$ fpm show a $(\Delta t_c/\Delta t_o)_{Kp} = 0.70$ (approximately) at K_p. Therefore, $C_2 = 2.43 \times 0.70\ K_p^2 = 1.7\ K_p^2$.

Ruden[6] obtained data on a heated jet with a $\Delta t_o = 31$ F deg (2.83 in. round nozzle). His axial velocity equation was,

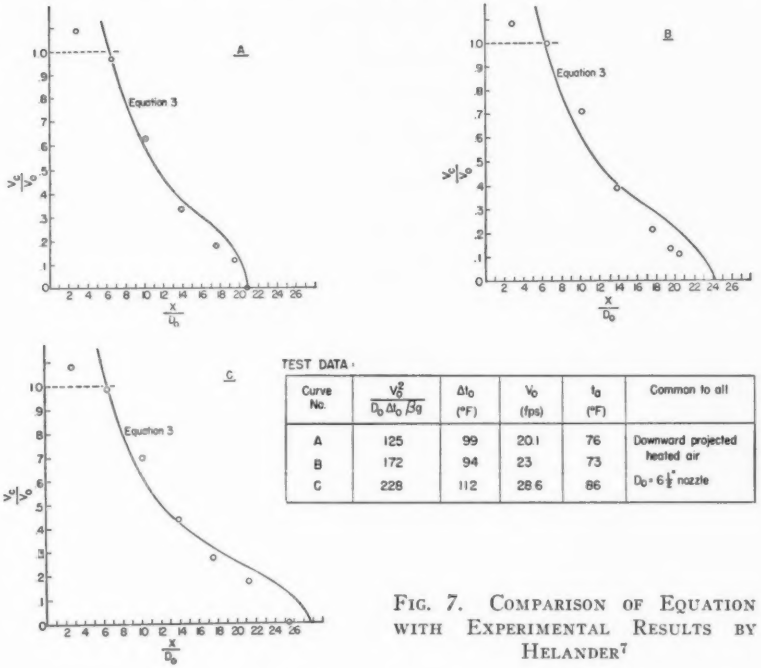
$$V_c/V_o = 6.45/[(X_o/D_o) + 0.66]$$

and his axial temperature difference equation was,

$$\Delta t_c/\Delta t_o = 5.86/[(X_o/D_o) + 1.41]$$

From Ruden's equations the $(\Delta t_c/\Delta t_o)_{Kp} = 0.814$ at K_p; therefore, $C_2 = 1.98\ K_p^2$.

These values of C_2 for non-isothermal jets are comparable to the value of C_2 derived for the isothermal jet, *i.e.*, $C_2 = 2K_p^2$.



TEST DATA

Curve No.	$\frac{V_o^2}{D_o \Delta t_o \beta g}$	Δt_o (°F)	V_o (fps)	t_a (°F)	Common to all
A	125	99	20.1	76	Downward projected heated air
B	172	94	23	73	
C	228	112	28.6	86	$D_o = 6\frac{1}{2}$" nozzle

FIG. 7. COMPARISON OF EQUATION WITH EXPERIMENTAL RESULTS BY HELANDER[7]

By combining Equations 5 and 3 the centerline temperature difference as a function of distance becomes

$$\Delta t_c/\Delta t_o = \frac{\frac{C_2(1 + b/a)}{4K_p X/D_o}}{\left[1 \pm \frac{3}{8} \frac{C_2(a/b + 1)}{K_p^3} \frac{D_o \Delta t_o \beta g}{V_o^2} \left(\frac{X}{D_o}\right)^2\right]^{\frac{1}{3}}} \quad \ldots\ldots (6)$$

Equations 5 and 6 do not apply in such a terminal region of a jet where the heat flows radially outward as the velocities in the X direction are reduced to zero (*i.e.*, heated air projected downward and chilled air projected upward) (see Fig. 5). In this region Equations 5 and 6 give increasing values for $\Delta t_c/\Delta t_o$ with an increase in X/D_o which is, of course, not possible.

Further equations, for determining the velocity and temperature envelopes, are derived in the Appendix. These envelopes are defined as the loci of all points of equal velocity or equal temperature-difference, thereby defining the shape of the stream.

COMPARISON WITH TEST DATA

In Fig. 7, Equation 3 is compared with test data on heated air projected downward. The distance from the outlet face to the apparent point source was

assumed as 2.85 diameters for the test points and the value for the overall effective turbulent Prandtl number was taken as 0.7. For a C_2 equal to $2K_p^2$, the agreement between equation and test data is good when a value of K_p equal to 6.5 is used. A value of K_p equal to 6.5 in conjunction with an A of 2.85 is somewhat lower than anticipated.

Equation 3 indicates that for chilled jets projected upward and heated jets projected downward, the conditions for which the sign is minus, there is a distance X/D_o for which V_c/V_o becomes zero. This distance, termed *maximum*

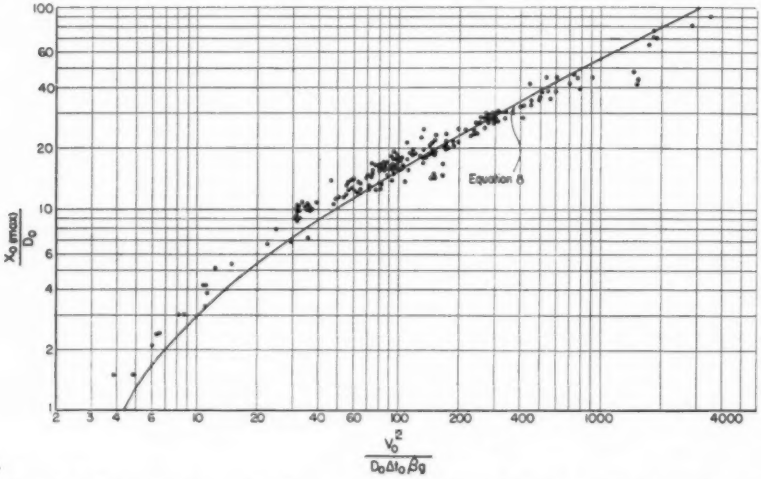


FIG. 8. COMPARISON OF EQUATION 8 WITH TEST POINTS BY HELANDER[7] FOR MAXIMUM TRAVEL OF JET

downward blow, for heated jets has been reported by Helander[7], who has obtained extensive data on this boundary condition. When $V_c/V_o = 0$, $X/D_o = X_{max}/D_o$ and Equation 3 becomes,

$$\frac{8\left(\frac{V_o^2}{D_o \Delta t_o \beta g}\right) K_p^2}{3C_2(a/b + 1)} = \left(\frac{X_{max}}{D_o}\right)^2 \quad \ldots\ldots\ldots\ldots (7)$$

If the same values for A, N_{Prt}, C_2 and K_p are assumed as in Fig. 7, we have,

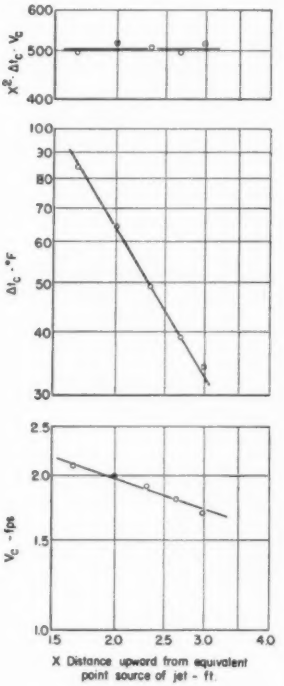
$$3.4\left(\frac{V_o^2}{D_o \Delta t_o \beta g}\right) = \left[\left(\frac{X_{o(max)}}{D_o}\right) + 2.85\right]^2 \quad \ldots\ldots\ldots\ldots (8)$$

This equation is compared with Helander's test data in Fig. 8.

In Fig. 9 data obtained by Schmidt[8] are plotted on an upward projected heated jet produced by natural convection. For this jet the centerline velocity varies as $X^{-1/3}$ and the centerline temperature difference as $X^{-5/3}$. Equation 3 would indicate this one-third slope when the term on the right side of the plus or minus sign is large compared to the left hand term. The product $(\Delta t_c V_c X^2)$ computed from Schmidt's data is also plotted in Fig. 9. Note that the product

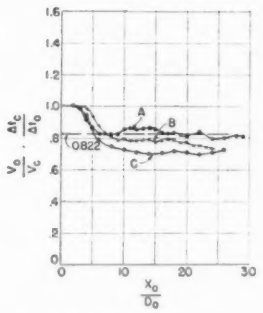
is relatively constant and independent of distance X, as pointed out by Equation 5.

For horizontally projected non-isothermal jets for which the trajectory is reasonably horizontal, the ratio of momentum to heat flow should be a constant. Under these conditions Equation 28 in the Appendix was derived. This equation is $(V_o/V_c)\ (\Delta t_c/\Delta t_o) = \frac{1}{2}(1 + b/a)$. When N_{Prt} is taken as 0.7, $(V_o/V_c)\ (\Delta t_c/\Delta t_o) = 0.822$. The value 0.822 is compared with the product $(V_o/V_c)\ (\Delta t_c/\Delta t_o)$, computed from test data obtained by Corrsin and Uberoi in Fig. 10.



(From data reported by Schmidt[8])

FIG. 9. CHARACTERISTICS OF A HEATED JET IN NATURAL CONVECTION $D_o = 2.17$ IN., $A = 2.0$ DIAMETERS



Note: D_o = 1-in. round nozzle.
Curve A is for Δt_o = 540 F deg.
Curve B is for Δt_o = 27 F deg.
Curve C is for Δt_o = 306 F deg.

FIG. 10. THE PRODUCT $(V_o/V_c)\ (\Delta t_c/\Delta t_o)$ *vs.* DISTANCE FROM OUTLET AS SHOWN BY DATA OF CORRSIN AND UBEROI[5]

According to Helander's unpublished results (Progress Report, Downward Projection of Heated Air, January 6, 1951) an equation of the form

$$\Delta t_c/\Delta t_o = 6/[(X_o/D_o) + 1]$$

roughly correlated his data on centerline temperature differences for values of

$$[D_o \Delta t_o \hat{\rho} g / V_o^2]$$

ranging from 0.0024 to 0.0013. Note that the distance from the nozzle face to the apparent point source for this temperature difference equation is estimated

as unity. Equation 6 is plotted (also see Equation 9) in Fig. 11 for a

$$[D_o \Delta t_o \beta g / V_o^2]$$

equal to 0.001370 and compared with corresponding test data. The same values for N_{Prt}, C_2, and K_p are assumed as in Fig. 7 for which Equation 6 becomes,

$$\Delta t_c / \Delta t_o = \frac{\frac{5.35}{X/D_o}}{\left[1 \pm 0.294\left(\frac{D_o \Delta t_o \beta g}{V_o^2}\right)\left(\frac{X}{D_o}\right)^2\right]^{\frac{1}{3}}} \quad \ldots\ldots\ldots (9)$$

Note the increasing values for $\Delta t_c/\Delta t_o$ in Fig. 11 in the region where radial outflow begins. It must be remembered that Equation 6 is valid only for decreasing values of $\Delta t_c/\Delta t_o$. As shown in Fig. 11, Equation 9 approximates Helander's correlation equation.

CONCLUSIONS AND RECOMMENDATIONS

Equations are given in this paper to provide a basis for exploring the performance of all non-isothermal air jets, heated or cooled, discharging vertically

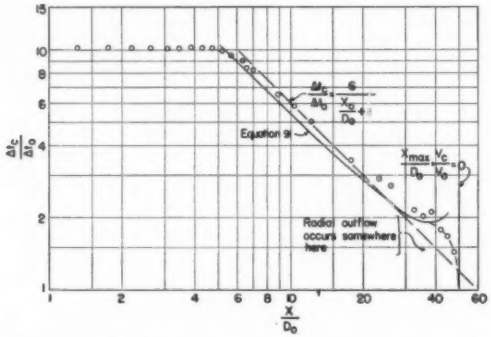


FIG. 11. COMPARISON OF EQUATION 9 WITH CENTERLINE TEMPERATURE DIFFERENCES OBTAINED BY HELANDER.[7] FOR VALUE OF $D_o \Delta t \beta g / V_o^2 = 0.00137$

into undisturbed space, either upward or downward. Approximate values of the constants in these equations are given, based on theory and available experimental data, but suggestions are made for additional experimental work to verify and refine these constants. Equations 3 and 6 give the temperature and velocity trajectory of vertically projected non-isothermal jets in which buoyancy, heat diffusion, and momentum play an important part. Equations 1 and 2 are the familiar velocity equations covering isothermal jets.

Simplifying assumptions have been made, as in all analytical treatments of the jet problem, and further experiment is necessary to indicate the seriousness of actual deviations from the assumed conditions. If, for instance, the differences in density between the jet medium and the ambient air are too great, the equations might give valid results if a correction involving the absolute temperature ratio is applied. The validity of the assumptions in regions of low velocities may

be questionable. Each assumption should be individually checked in the laboratory in order to accurately determine its limitations.

Helander and his associates, in their extensive work on heated jets projected downward, have called attention to some of the differences between the actual conditions and those assumed for analysis. One of these is the radial escape of warm stream air in the terminal zone of a hot stream projected downward. This escaping warm air then rises and causes a change in ambient conditions for the upper part of the jet. The terminal zone and also the edge of the jet are zones of marked instability, with definite surges and fluctuations, so that the jet envelope is very difficult to define or to determine experimentally.

The equations herewith are recommended for engineering use, pending their refinement by more rigorous experimental work. The derived equations are not to be considered exact until full experimental verification is accomplished and any limitations on the empiricism determined.

ACKNOWLEDGMENTS

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APPENDIX

Momentum Analysis

Experimenters such as Albertson[A-1], Taylor[A-2], Keagy[A-3], Pai[A-4], and Tuve[1] have demonstrated that the Gauss error-function or normal probability type curve is an

accurate representation of the cross-sectional velocity profile of a free jet. We can then let an equation of the form,

$$V = V_c\, e^{-ar^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

define the shape of the velocity profile. The factor a can be called a shape factor since its value determines the width of the profile for a given value of centerline velocity, V_c.

The stream momentum, M_X, at any arbitrary distance, X, from an apparent point source is,

$$M_X = 2\pi\rho \int_{r=0}^{r=\infty} V^2 r\, dr \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

where ρ is assumed constant throughout the field of flow. Fig. A-1 is the diagram to which Equation 11 is applied. If the function of V in Equation 10 is substituted into Equation 11, the integration can be accomplished as follows:

$$M_X = 2\pi\rho V_c^2 \int_{r=0}^{r=\infty} e^{-2ar^2} r\, dr \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (12)$$

Let $u = -2ar^2$ therefore $du = -4ardr$

Since $d(e^u) = e^u du$

$$d(e^{-2ar^2}) = -4ar\, dr e^{-2ar^2}$$

and

$$\frac{d(e^{-2ar^2})}{-4a} = e^{-2ar^2} r dr \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (13)$$

The substitution of Equation 13 into Equation 12 gives

$$M_X = 2\pi\rho\, V_c^2 \int_{r=0}^{r=\infty} \frac{d(e^{-2ar^2})}{-4a} = \frac{2\pi\rho V_c^2}{-4a} \left[e^{-2ar^2} \right]_{r=0}^{r=\infty}$$

and,

$$M_X = \frac{\pi\rho V_c^2}{2a} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (14)$$

Equation 14 is basic equation relating the momentum of the jet to the centerline velocity and the shape of the jet. This equation will be used in conjunction with the preceding analysis on non-isothermal jets.

HEAT FLOW ANALYSIS FOR NON-ISOTHERMAL JETS

If it is assumed that the temperature difference profile in a non-isothermal jet can also be defined by a normal probability type curve, then

$$\Delta t = \Delta t_c e^{-br^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (15)$$

where b is the shape factor for the temperature difference profile. In this case, b will not have the same value as a due to the difference in the rates of momentum and heat diffusion. It has been shown experimentally that the temperature profile is flatter or more spread out than the velocity profile at a given stream section.

If the heat flow in a non-isothermal jet is measured above or below the ambient temperature as would be the case for a heated or chilled jet, it can be stated that the heat flow in the free stream is equal to the heat flow at the outlet, or

$$Q_o = Q_X \quad \ldots\ldots\ldots \quad \text{(conservation of thermal energy)}$$

If the heat flow equation is applied to a free stream having a temperature difference and a velocity profile as shown in Fig. A-2,

$$Q_X = \gamma C_p 2\pi \int_{r=0}^{r=\infty} \Delta t V r dr \quad \ldots \ldots \ldots \ldots \quad (16)$$

where γ is assumed constant throughout the jet. This assumption is valid if the temperature differences are not too great.

If Equations 10 and 15 are substituted into Equation 16, the heat flow equation takes the following form:

$$Q_X = \gamma C_p 2\pi \int_{r=0}^{r=\infty} \Delta t_c V_c e^{-br^2} e^{-ar^2} r dr$$

or,

$$Q_X = \gamma C_p 2\pi \Delta t_c V_c \int_{r=0}^{r=\infty} e^{-r^2(a+b)} r dr$$

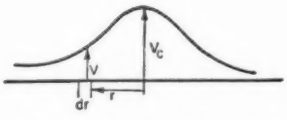


FIG. A-1. VELOCITY CROSS-SECTION OF JET

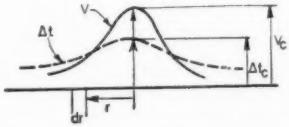


FIG. A-2. VELOCITY AND TEMPERATURE DIFFERENCE CROSS-SECTION OF JET

The integration may be accomplished as follows:

Let $u = -r^2(a + b)$ therefore $du = -2(a + b)\, r\, dr$

Since $d(e^u) = e^u\, du$

$$d(e^{-r^2(a+b)}) = -2(a + b)\, r\, dr\, e^{-r^2(a+b)}$$

and

$$\frac{d(e^{-r^2(a+b)})}{-2(a + b)} = e^{-r^2(a+b)} r\, dr$$

therefore

$$Q_X = \frac{\gamma C_p 2\pi \Delta t_c V_c}{-2(a + b)} \int_{r=0}^{r=\infty} d(e^{-r^2(a+b)})$$

$$Q_X = \frac{\gamma C_p 2\pi \Delta t_c V_c}{-2(a + b)} \left[e^{-r^2(a+b)} \right]_{r=0}^{r=\infty}$$

and

$$Q_X = \frac{\gamma C_p \pi \Delta t_c V_c}{a + b} \quad \ldots \ldots \ldots \ldots \quad (17)$$

where Q_X is also equal to Q_o.

Equation 17 is a basic equation relating the stream heat flow at a given distance X, to the centerline velocity and temperature difference, and to the shape of the temperature difference and velocity profiles.

BUOYANT FORCE ANALYSIS FOR NON-ISOTHERMAL JETS

In order to investigate the flow behavior of vertically projected non-isothermal jets, the buoyant forces and the resulting changes in stream momentum must be taken into account. The buoyant force at any cross-section of a non-isothermal jet can be determined from the assumed temperature difference profile, *i.e.*, Equation 15. The buoyant force per unit volume for a gas is $\gamma\beta\Delta t$ where β is the coefficient of expansion and is equal to $1/T_a$ for a perfect gas.

The buoyant force F at a given stream cross section dX in length is,

$$F = dX\beta\gamma 2\pi \int_{r=0}^{r=\infty} \Delta t r dr \quad \ldots\ldots\ldots\ldots (18)$$

and Fig. A-3 is the applicable diagram.

Substituting Equation 15, the temperature difference profile equation, into Equation 18, gives,

$$F = \Delta t_c \beta\gamma 2\pi dX \int_{r=0}^{r=\infty} e^{-br^2} r dr$$

The integration may be accomplished as follows:

Let $u = -br^2$ therefore $du = -2\,br dr$

Since $d(e^u) = e^u du$

$$d(e^{-br^2}) = -2\,br\,dr\,e^{-br^2}$$

and

$$\frac{d(e^{-br^2})}{-2b} = e^{-br^2} r dr$$

Therefore

$$F = \frac{\Delta t_c \beta\gamma 2\pi dX}{-2b} \left[e^{-br^2} \right]_{r=0}^{r=\infty}$$

and

$$F = \frac{(\Delta t_c \beta\gamma\pi) dX}{b} \quad \ldots\ldots\ldots\ldots\ldots (19)$$

Equation 19 relates the buoyant force F for a jet dX in length to the centerline temperature difference and the shape of the temperature difference profile.

SHAPE FACTORS a AND b AS A FUNCTION OF X

Assuming a constant angle of expansion, α_v, for the velocity profile, the ratio V_1/V_c measured along the expansion line becomes independent of X. The ratio V_1/V_c, could be taken as one half as in Fig. 2. The velocity profile Equation 10 in terms of angle of expansion is,

$$\frac{V_1}{V_c} = e^{-a\eta^2} = e^{-a(\tan\alpha_v)^2 X^2}$$

Differentiating with respect to X gives,

$$d\left(\frac{V_1}{V_c}\right) = 0 = d\left(e^{-a(\tan\alpha_v)^2X^2}\right)$$

Letting $u = -a(\tan \alpha_v)^2X^2$

$$du = (\tan \alpha_v)^2 [-a\ 2X\ dX - X^2\ da]$$

Since $d(e^u) = e^u\ du$

the differentiated equation becomes,

$$0 = e^{-a(\tan\alpha_v)^2X^2} [-a\ 2X\ dX - X^2\ da] (\tan \alpha_v)^2$$

or $X^2\ da = -a\ 2X\ dX$

and $-\dfrac{da}{a} = 2\dfrac{dX}{X}$

integrating $-\ln a = 2 \ln X + C_2''$

where C_2'' is a constant of integration

also $\ln X^2a = C_2'$ . . . . . (20)

$$\text{or} \quad a = \frac{C_2}{X^2}$$

A similar analysis on the temperature difference profile gives,

$$b = C_1/X_1^2 \text{ . . . . . (21)}$$

Dividing Equation 21 by Equation 20 gives,

$$\frac{b}{a} = \frac{C_1/X_1^2}{C_2/X^2} = \frac{C_1}{C_2}\left(\frac{X}{X_1}\right)^2$$

which indicates that the ratio of the shape factors b and a is a constant at a large enough distance from the outlet where X/X_1 approaches unity.

ADDITIONAL NOMENCLATURE FOR APPENDIX

A_o = Outlet area, square feet.
C_p = constant pressure specific heat Btu per pound per degree Fahrenheit.
C = constant of integration.
C_1 = constant of integration.
F = buoyant force, pounds.
M_o = outlet momentum, pounds.
M_X = momentum of the jet at distance X, pounds.
Q_o = heat flow or enthalpy flow of the jet at the outlet measured above ambient temperature, Btu per second.
Q_X = heat flow or enthalpy flow of the jet at distance X measured above ambient temperature, Btu per second.
β = coefficient of expansion. For a perfect gas, $\beta = \dfrac{1}{T_a}$ where T_a is the absolute temperature of the ambient air.
γ = density in pounds per cubic foot.
ρ = density in slugs per cubic foot.
δ_m = momentum displacement diameter feet (defined as used in Equation 24 Appendix).
δ_t = heat flow displacement diameter feet (defined as used in Equation 26 Appendix).

The ratio b/a can be related to the ratio $\tan \alpha_v / \tan \alpha_t$ if the two profile equations are divided as follows,

$$\frac{\Delta t_1 / \Delta t_c}{V_1 / V_c} = 1 = \frac{e^{-b(\tan \alpha_t)^2 X_1^2}}{e^{-a(\tan \alpha_v)^2 X^2}} = e^{-b(\tan \alpha_t)^2 X_1^2 + a(\tan \alpha_v)^2 X^2}$$

Since $e^0 = 1$

$$b\ (\tan \alpha_t)^2\ X_1^2 = a(\tan\ \alpha_v)^2\ X^2$$

or

$$\frac{b}{a} = \left(\frac{\tan\ \alpha_v}{\tan\ \alpha_t}\right)^2 \left(\frac{X}{X_1}\right)^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ (22)$$

Data obtained by Boelter and Cleeves[A-5] on very high temperature heated jets ($\Delta t_o = 1100$ F to 1600 F) show a value of $\tan \alpha_v / \tan \alpha_t$ equal to approximately 0.8. This gives a value of $b/a = (0.8)^2 = 0.64$, if X is taken as equal to X_1.

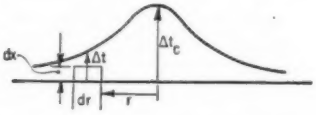


FIG. A-3. TEMPERATURE DIFFERENCE CROSS-SECTION OF JET

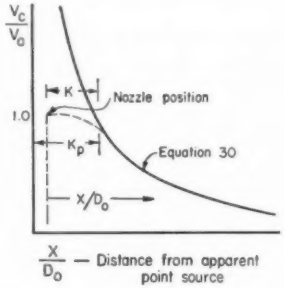


FIG. A-4. CENTERLINE VELOCITY TRAJECTORY OF ISOTHERMAL JET

SHAPE FACTORS a AND b AS A FUNCTION OF THE PRANDTL NUMBER

Corrsin and Uberoi[5] obtained an equation which related a momentum diameter and a thermal diameter of a non-isothermal jet to the over-all effective turbulent Prandtl number.

This equation is,

$$\left(\frac{\delta_t}{\delta_m}\right)^2 = \frac{1}{2}\left(1 + \frac{1}{N_{Pr_t}}\right) \quad \ldots\ldots\ldots\ldots\ldots\ (23)$$

Corrsin and Uberoi also found that the average of the effective turbulent Prandtl number in a round jet was equal to the laminar value, within the accuracy of their measurements. For a Δt_o equal to 15 C and 300 C they obtained values for the Prandtl number of 0.701 and 0.690, respectively. Nottage[3] also reports a turbulent Prandtl number of 0.70 on chilled jets. Forstall and Shapiro[4] recommend a value of 0.70 $\pm$10 percent for the turbulent Prandtl number. Their studies were made on high velocity heated jets. The momentum diameter, δ_m,* and the thermal diameter δ_t,* of a non-isothermal jet are defined by Equations 24 and 26 as follows:

Momentum:

$$2\pi\rho \int_{r=0}^{r=\infty} V^2 r\, dr = \delta_m^2 \frac{\pi}{4} V_c^2 \rho = M_X \quad \ldots\ldots\ldots\ldots\ (24)$$

*Physically δ_m is the diameter of a jet having a rectangular velocity profile and whose total axial momentum is the same as in the actual jet concerned. Similar reasoning is involved for the physical significance of δ_t.

Since

$$M_X = \frac{\pi \rho V_c^2}{2a} \qquad \text{(Equation 14)}$$

then

$$\frac{\pi \rho V_c^2}{2a} = \delta_m^2 \frac{\pi}{4} V_c^2 \rho$$

or

$$\frac{2}{a} = \delta_m^2 \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (25)$$

Heat Flow:

$$\gamma C_p 2\pi \int_{r=0}^{r=\infty} \Delta t V r \, dr = \delta_t^2 \frac{\pi}{4} V_c \Delta t_c \gamma C_p = Q_X \quad \ldots \ldots \ldots \quad (26)$$

Since

$$Q_X = \frac{\gamma C_p \pi \Delta t_c V_c}{a + b} \qquad \text{(Equation 17)}$$

then,

$$\frac{\gamma C_p \pi \Delta t_c V_c}{a + b} = \delta_t^2 \frac{\pi}{4} V_c \Delta t_c \gamma C_p$$

or

$$\frac{4}{a + b} = \delta_t^2 \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (27)$$

Combining Equations 23, 25, 27, we have,

$$\frac{\delta_t^2}{\delta_m^2} = \frac{\dfrac{4}{a + b}}{\dfrac{2}{a}} = \frac{1}{2}\left(1 + \frac{1}{N_{Prt}}\right)$$

or

$$\frac{4}{1 + \dfrac{1}{N_{Prt}}} - 1 = \frac{b}{a} \quad \ldots \ldots \ldots \ldots \quad (28)$$

For a Prandtl number of 0.7, $b/a = 0.65$, which checks with the results obtained from Equation 22 using data on angles of expansion α_v and α_t.

For horizontally projected non-isothermal jets for which the trajectory is reasonably horizontal, the ratio of momentum to heat flow should be a constant; therefore, combining Equations 14 and 17,

$$\frac{M_o}{Q_o} = \frac{M_X}{Q_X} = \frac{\dfrac{\pi \rho V_c^2}{2a}}{\dfrac{\gamma C_p \pi \Delta t_c V_c}{a + b}}$$

$$\frac{V_o^2 \dfrac{\pi D_o^2}{4} \rho}{\dfrac{\pi D_o^2}{4} V_o \gamma C_p \Delta t_o} = \frac{\dfrac{\pi \rho V_c^2}{2a}}{\dfrac{\gamma C_p \pi \Delta t_c V_c}{a + b}}$$

Simplifying

$$\frac{V_o\,\Delta t_c}{\Delta t_o\,V_c} = \frac{1}{2}\left(1 + \frac{b}{a}\right)$$

For a Prandtl number of 0.7, $\left(1 + \frac{b}{a}\right) = 1.645$, therefore

$$\frac{V_o\,\Delta t_c}{V_c\,\Delta t_o} = 0.822 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (29)$$

In Fig. 10 are plotted values of $\frac{V_o\,\Delta t_c}{V_c\,\Delta t_o}$ for horizontally projected heated jets of three different supply temperature differences.

DERIVATION OF EQUATIONS

Conservation of momentum is usually assumed for an isothermal jet analysis; therefore, Equation 14 can be equated to the momentum at the outlet as follows:

$$M_X = \frac{\pi\rho V_c^2}{2a} = M_o$$

where

$$M_o = \frac{\pi D_o^2}{4} V_o^2\rho \text{ and } a = \frac{C_2}{X^2} \qquad \text{(Equation 20)}$$

Therefore

$$\frac{\pi D_o^2}{4} V_o^2\rho = \frac{\pi\rho V_c^2}{2a} = \frac{\pi\rho V_c^2 X^2}{2C_2}$$

and

$$\frac{V_c}{V_o} = \sqrt{\frac{C_2}{2}}\,\frac{D_o}{X} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (30)$$

Equation 30 is an equation of a hyperbola as shown in Fig. A-4. The constant C_2 may be evaluated from available experimental data on isothermal jets by considering the conditions existing in the constant velocity zone near the outlet. When $V_c/V_o = 1$, $X/D_o = K_p$ and $C_2 = 2K_p^2$. Fig. A-5 shows values of K_p obtained by Nottage[3]. Davies[A-6] gives the following equation for a isothermal jet:

$$\frac{V_c}{V_o} = \frac{8.4}{\frac{X_o}{D_o} + 2} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (31)$$

where X_o/D_o is the distance in diameters measured from the outlet and 2 is the distance in diameters from the outlet face to the apparent point source (Note: $X_o/D_o + 2 = X/D_o$). According to Davies' equation K_p has a value of 8.4; which is lower than the values shown in Fig. A-5. Davies' apparent point source distance of $2D_o$ is also lower than the values given in Fig. A-5. Other experimenters have found the apparent point source to lie on the face of the outlet.

For non-isothermal jets the momentum changes with respect to X due to the buoyant forces; therefore M_X is not equal to M_o. An equation in the form of $F = \pm\, d(M_X)$ can be written, and expressions already derived for the buoyant force and momentum substituted. The plus or minus sign indicates an increase or decrease in momentum depending on the direction of the buoyant force with respect to the momentum.

Since

$$F = \pm\, d(M_X) \text{ then,}$$

$$\frac{(\Delta t_c \beta \gamma \pi)\, dX}{b} = \pm\, d\left(\frac{\pi \rho V_c^2}{2a}\right) \quad \ldots\ldots\ldots\ldots \quad (32)$$

where

$$\frac{(\Delta t_c \beta \gamma \pi)\, dX}{b} = F \qquad \text{(Equation 19)}$$

$$\frac{\pi \rho V_c^2}{2a} = M_X \qquad \text{(Equation 14)}$$

and

$$a = \frac{C_2}{X^2} \qquad \text{(Equation 20)}$$

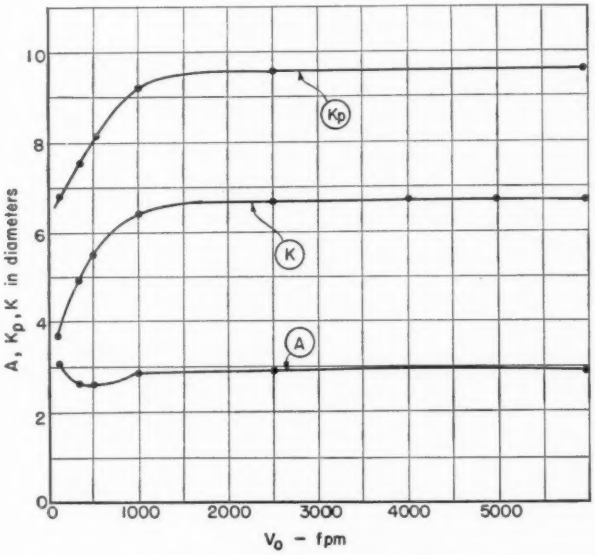


FIG. A-5. LENGTHS OF THE APPARENT CONSTANT VELOCITY ZONE AND DISTANCE TO POINT SOURCE FOR A 6-IN. DIAMETER NOZZLE. DATA OBTAINED BY NOTTAGE[2]

Since the heat flow, Q_X, is constant for the non-isothermal jet, Δt_c can be eliminated from Equation 32 as follows:

Since

$$Q_o = Q_X = \frac{\gamma C_p \pi \Delta t_c V_c}{a + b} \qquad \text{(Equation 17)}$$

$$Q_o = \frac{\pi D_o^2}{4}\, V_o \Delta t_o \gamma C_p$$

therefore

$$\Delta t_c = \frac{D_o^2 V_o \Delta t_o (a+b)}{4 V_c} \quad \ldots\ldots\ldots\ldots (33)$$

Substituting Equation 33 into Equation 32,

$$\frac{\pi D_o^2}{4} V_o \Delta t_o \beta\gamma \left(\frac{a}{b}+1\right) \frac{dX}{V_c} = \pm d\left(\frac{\pi \rho V_c^2 X^2}{2C_2}\right)$$

$$gC_2 \frac{D_o^2}{2} V_o \Delta t_o \beta \left(\frac{a}{b}+1\right) \frac{dX}{V_c} = \pm d\left(V_c^2 X^2\right)$$

Differentiating the right side of the equation,

$$gC_2 \frac{D_o^2}{2} V_o \Delta t_o \beta \left(\frac{a}{b}+1\right) \frac{dX}{V_c} = \pm \left(V_c^2\, 2X\, dX + X^2\, 2V_c\, dV_c\right)$$

Let

$$Z = gC_2 \frac{D_o^2}{2} V_o \Delta t_o \beta \left(\frac{a}{b}+1\right), \text{ which is a constant;}$$

therefore

$$\frac{Z}{2} \frac{dX}{V_c} = V_c^2 X\, dX + X^2 V_c\, dV_c$$

and

$$\pm \frac{Z}{2} dX = V_c^2 X\, dX + X^2 V_c^2\, dV_c \quad \ldots\ldots\ldots\ldots (34)$$

The solution of this differential equation is

$$V_c^3 = \frac{C}{X^3} \pm \frac{3}{4} \frac{Z}{X} \quad \ldots\ldots\ldots\ldots (35)$$

where C is a constant of integration.

The constant C can be evaluated by considering the condition where Δt_o is equal to zero (isothermal). When $\Delta t_o = 0$, then Z is equal to zero and Equation 35 becomes

$$V_c^3 = \frac{C}{X^3} \qquad \text{or} \qquad V_c = \frac{C^{1/3}}{X} \quad \ldots\ldots\ldots\ldots (36)$$

which is an equation of a hyperbola similar to Equation 30 for an isothermal jet. When $V_c = V_o$, X becomes $K_p D_o$ (see Fig. A-4); therefore,

$$C = (V_o K_p D_o)^3 \quad \ldots\ldots\ldots\ldots (37)$$

and Equation 35 becomes

$$V_c^3 = \left(\frac{V_o K_p D_o}{X}\right)^3 \pm \frac{3}{4} \frac{Z}{X}$$

and by substituting the values for Z and rearranging terms,

$$\left(\frac{V_c}{V_o}\right)^3 = \left(\frac{K_p}{X/D_o}\right)^3 \pm \frac{3}{8} C_2 \left(\frac{a}{b}+1\right)\left(\frac{D_o \Delta t_o \beta g}{V_o^2}\right) \frac{1}{X/D_o} \quad \ldots\ldots (38)$$

It has been shown that for an isothermal jet, C_2 is equal to $2K_p^2$ (see Equation 30 and its discussion), and it may be intuitively expected that C_2 for a non-isothermal jet may approximate this value. If conservation of thermal energy is assumed in the X direction or, in other words, along the trajectory of the jet, an expression by

which C_2 can be evaluated can be derived. If the heat flow in the free jet, Q_x (Equation 17), is equated to the heat flow from the nozzle,

$$\frac{\pi D_o^2}{4} V_o \gamma C_p \Delta t_o = \frac{\gamma C_p \pi \Delta t_o V_c}{a+b} = \frac{\gamma C_p \pi \Delta t_c V_c}{a\left(1+\frac{b}{a}\right)}$$

substituting $a = C_2/X^2$ into the equation and simplifying,

$$\frac{C_2\left(1+\frac{b}{a}\right)}{4} = \left(\frac{\Delta t_c}{\Delta t_o}\right)\left(\frac{V_c}{V_o}\right)\left(\frac{X}{D_o}\right)^2 \quad \ldots\ldots\ldots\ldots (39)$$

The constant C_2, in Equation 39 may be evaluated by considering data on any non-isothermal jet regardless of its trajectory. When X/D_o is taken as the apparent core length, $K_{p,n}$ for the constant velocity zone, V_c/V_o is equal to unity; therefore, Equation 39 becomes

$$\frac{C_2\left(1+\frac{b}{a}\right)}{4} = \left(\frac{\Delta t_c}{\Delta t_o}\right)_{K_{p,n}}\left(K_{p,n}\right)^2 \quad \ldots\ldots\ldots\ldots (40)$$

It is well known that the rate of spread of a non-isothermal, turbulent jet is affected by the difference in density of the jet fluid and of the receiving medium. This difference in rate of spread in turn effects the apparent core length as shown in Fig. 4. For example, a heated air jet with a Δt_o of 100 F discharged into ambient air at 80 F ($\Delta t_o \beta = \frac{100}{80+460} = 0.185$) will have, according to Fig. 4, a core length $K_{p,n}$ which is equal to 97 percent of the core length of an isothermal jet K_p. In keeping with the assumption that the temperature differences involved are not too great, $K_{p,n}$ can be taken as equal to K_p. Equation 40 becomes,

$$\frac{C_2\left(1+\frac{b}{a}\right)}{4} = \left(\frac{\Delta t_c}{\Delta t_o}\right)_{K_p}\left(K_p\right)^2 \quad \ldots\ldots\ldots\ldots (41)$$

For a turbulent Prandtl number of 0.7, the quantity $\left(1+\frac{b}{a}\right)$ is equal to 1.645 according to Equation 28. Therefore, Equation 41 becomes

$$C_2 = 2.43\left(\frac{\Delta t_c}{\Delta t_o}\right)_{K_p}\left(K_p\right)^2 \quad \ldots\ldots\ldots\ldots (42)$$

From data obtained by experimenters on horizontally projected non-isothermal jets, the temperature differences $(\Delta t_c/\Delta t_o)$ can be evaluated at K_p, and by this means C_2 can be determined. The following values were obtained for C_2:

SOURCE	OUTLET CONDITIONS	$(\Delta t_c/\Delta t_o)K_p$	C_2
Corrsin and Uberoi[5] (see Fig. 6.)	Δt_o = 55F, heated, 1 in. round nozzle	0.75	$1.82K_p^2$
Nottage[8]	Δt_o = 39F, chilled, V_o = 500fpm, 6. in round nozzle	0.70	$1.7K_p^2$
Ruden[6]	Δt_o = 31F, heated, 2.83 in. round nozzle	0.814	$1.98K_p^2$

These values for C_2 approximate the value of $C_2 = 2K_p^2$ obtained for an isothermal jet.

Equation 39 brings out an interesting point that the product $(\Delta t_c V_c X^2)$ should be a constant if Q_x is conserved in the X direction. In Fig. 9 the product $(\Delta t_c V_c X^2)$ is plotted against X for data obtained by Schmidt on an upward projected heated jet produced by natural convection. As can be seen the agreement is good.

If Equations 38 and 39 are combined and simplified we obtain an equation relating the centerline temperature difference to distance. The equation is,

$$\frac{\Delta t_c}{\Delta t_o} = \frac{\dfrac{C_2\left(1+\dfrac{b}{a}\right)}{4K_p X/D_o}}{\left[1 \pm \dfrac{3}{8}\dfrac{C_2}{K_p{}^3}\left(\dfrac{a}{b}+1\right)\left(\dfrac{D_o \Delta t_o \beta g}{V_o{}^2}\right)\left(\dfrac{X}{D_o}\right)^2\right]^{\frac{1}{3}}} \quad \ldots\ldots (43)$$

There are certain limitations to the use of Equations 43 and 39, near the terminal region of the jet trajectory for downward projected heated air and upward projected chilled air. As V_c is reduced to zero due to the opposing buoyant forces, the heat flow in the X direction, Q_x, can no longer be conserved. Helander[7] experimentally observed that in this region radial outflow of the jet air occurred, as is schematically shown in Fig. 5. The assumption that thermal energy is conserved in the X direction becomes inaccurate. Under these conditions Equation 39 and 43 indicate increasing values of $\Delta t_c/\Delta t_o$ with increasing X/D_o as is shown in Fig. 11. This is, of course, impossible and it must be kept in mind that the equations give valid results only for decreasing values of $\Delta t_c/\Delta t_o$. When heated air is projected downward and chilled air projected upward, the sign in Equation 38 is minus. If we let V_c/V_o equal zero, X/D_o becomes X_{max}/D_o, the maximum travel of the jet (see Fig. 5).

Equation 38 after simplifying becomes,

$$\frac{8\left(\dfrac{V_o{}^2}{D_o \Delta t_o \beta g}\right) K_p{}^3}{3C_2\left(\dfrac{a}{b}+1\right)} = \left(\frac{X_{max}}{D_o}\right)^2 \quad \ldots\ldots\ldots\ldots (44)$$

If the jet velocity and temperature difference profiles normal to the jet axis are desired, the assumed error-function Equations 10 and 15 can be applied. A velocity and a temperature difference envelope, defined as the locus of all points of equal velocity and temperature difference, can also be constructed from the error-function profiles. The necessary mathematical manipulations are as follows:

$V = V_c e^{-ar^2}$ (Equation 10),

$\Delta t = \Delta t_c e^{-br^2}$ (Equation 15)

$a = \dfrac{C_2}{X^2}$ (Equation 20) $\qquad b = \dfrac{C_1}{X_1{}^2}$ (Equation 21)

Since X is approximately equal to X_1, Equations 20 and 21 can be combined to obtain,

$$C_1 = C_2 \frac{b}{a}$$

By substitution, the profile equations become,

$$V = V_c e^{-C_2\left(\frac{r}{X}\right)^2} \quad \text{and} \quad \Delta t = \Delta t_c e^{-C_2\left(\frac{b}{a}\right)\left(\frac{r}{X}\right)^2}$$

Since C_2 is assumed equal to $2K_p^2$ for a non-isothermal jet substitution gives

$$V = V_c e^{-2\left(\frac{K_p r}{X}\right)^2} \quad \text{and} \quad \Delta t = \Delta t_c e^{-2\left(\frac{b}{a}\right)\left(\frac{K_p r}{X}\right)^2}$$

If these profile equations are substituted into Equations 38 and 43, equations by which temperature difference and velocity envelopes can be determined are obtained. Substitution gives the following:

$$\left(\frac{V}{V_o}\right)^2 e^{3C_2\left(\frac{r}{X}\right)^2} = \left(\frac{K_p}{X/D_o}\right)^2 \pm \frac{3}{8} C_2\left(\frac{a}{b}+1\right)\left(\frac{D_o \Delta t_o \beta g}{V_o^2}\right)\frac{1}{X/D_o} \quad \ldots (45)$$

and

$$\left(\frac{\Delta t}{\Delta t_o}\right) e^{C_2\left(\frac{b}{a}\right)\left(\frac{r}{X}\right)^2} = \frac{\frac{C_2\left(1+\frac{b}{a}\right)}{4K_p X/D_o}}{\left[1 \pm \frac{3}{8}\frac{C_2}{K_p^3}\left(\frac{a}{b}+1\right)\left(\frac{D_o \Delta t_o \beta g}{V_o^2}\right)\left(\frac{X}{D_o}\right)^2\right]^{\frac{1}{3}}} \quad \ldots (46)$$

From Equations 45 and 46 the locus of all points (defined geometrically by r and X) of equal V or Δt can be determined.

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DISCUSSION

LINN HELANDER, Manhattan, Kans. (WRITTEN): Professor Koestel's paper develops the characteristics of an idealized vertical jet under the premise that dynamic and geometric similarity will exist in the principal, or fully developed zone. His analysis provides suggestive insights into parametrical relationships and clues which should prove helpful to those of us who are engaged in experimental research on jets.

Professor Koestel's derived equations are basically in accord with analytical equations we have derived exploratively, using corresponding but not identical simplifying assumptions. At the present time we do not have the experimental data needed to either verify or disprove the practical validity of his or our equations. Professor Koestel has pointed out one limitation of his equations, namely, that they do not apply beyond the point in the jet where the mass rate of flow is a maximum. This point occurs where $X/D_o = 0.77\frac{X_{max}}{D_o}$. It is our opinion that this limitation restricts

not only the usefulness of his equation $\Delta T_c/\Delta T_o$, but that it also limits the application of his Equation 8 to long-throw jets, *i.e.*, jets that have a throw in excess of approximately 8 orifice diameters. For example, with a value of X_{max}/D_o of 6, there would exist a terminal zone equal to 1.4 orifice diameters over which his equations do not apply and a primary zone of 4.7 orifice diameters ($K_p = 6.5$) over which his equations likewise do not apply. This leaves, for this case, no section of the jet over which his equations are applicable. Professor Koestel's equation for throw shows rough agreement with the experimental data plotted in Fig. 8, but this should be regarded as the consequence of the mathematical form of his equations, not as the result of fundamental assumptions regarding the dynamics of a jet. One of our own equations developed semi-analytically: $X_{o,max}/D_o = \left(\frac{3V_o^2}{gD_o\Delta t_o\beta} - 9\right)^{\frac{1}{2}} - 1$; fits the data as well as Professor Koestel's Equation 8 does, but not as well as the empirical equations presented in our paper from which Fig. 8 was taken.

For long-throw jets wherein the ratio of the buoyancy force to the momentum force is small, we agree that the existence of similarity may be an acceptable, simplifying assumption. We are skeptical of the acceptability of the assumption where the ratio of the buoyancy force to the momentum force is not small. One condition that must be satisfied if dynamic and geometric similarity are to exist, is that the percentage rate of change of momentum flux with travel, $(dM/dX)/M$, evaluated for r/X held constant must be independent of r/X. That this requirement is satisfied by the distribution of buoyancy forces predicated by an error-function distribution of temperature difference is not apparent. We are skeptical of the applicability of the concept of similarity to jets as widely different in their flow pattern as are low-velocity, short-throw jets and high-velocity, long-throw jets, or to jets subjected to such widely different boundary conditions as are heated and cooled jets projected downward or upward.

In view of the order of agreement between Professor Koestel's equations and the available experimental data we would agree that his equations may be used for making practical estimates, but we would ourselves give preference to empirical equations where such equations are available.

On the basis of the fundamental assumptions employed by Professor Koestel, the following equation is easily derived for jets with either finite or infinite boundaries without defining the shape of the velocity and temperature profiles:

$$\left(\frac{V_c}{V_o}\right)^2 = \left(\frac{a_4 K_{p,n}}{X/D_o}\right)^3 \pm \frac{3}{16}\left(\frac{a_2}{a_1 a_3}\right)\left(\frac{g\, D_o \Delta t_o \beta}{V_o^2}\right)\left[1 - \left(\frac{K_{p,n}}{X/D_o}\right)^2\right]\left(\frac{1}{X/D_o}\right)$$

$$a_1 = \int_{(A)} \left(\frac{V}{V_c}\right)^2 \frac{d(r^2)}{X^2}$$

$$a_2 = \int_{(A)} \left(\frac{\Delta T}{\Delta T_c}\right) \frac{d(r^2)}{X^2}$$

$$a_3 = \int_{(A)} \left(\frac{V}{V_c}\right)\left(\frac{\Delta T}{\Delta T_c}\right) \frac{d(r^2)}{X^2}$$

$$a_4 = V_{c(p,n)}/V_o$$

(A) indicates that the integral is to be evaluated over the cross-sectional area of the jet.

The foregoing equations compare with Equation 3 of Professor Koestel's paper and take the following form under the assumptions he has employed:

$$\left(\frac{V_c}{V_o}\right)^2 = \left(\frac{a_4 K_{p,n}}{X/D_o}\right)^2 \pm \frac{3}{8}\, C_2\left(\frac{a}{b} + 1\right)\left(\frac{D_o \Delta T_o \beta g}{V_o^2}\right)\left[1 - \left(\frac{K_{p,n}}{X/D_o}\right)^2\right]\left(\frac{1}{X/D_o}\right)$$

By employing the foregoing equation together with Equations 3 and 7 of Professor Koestel's paper, the following relationship between $K_{p,n}$ and K_p is obtained for downward heated jets or upward cooled jets:

$$\left(\frac{a_4 K_{p,n}}{K_p}\right)^3 + \left(\frac{K_{p,n}}{X_{max}/D_o}\right)^2 = 1$$

This relationship might well be a subject for experimental investigation.

Professor Koestel has adequately explored the applicability of the concept of similarity to models of buoyant jets, and his results are both enlightening and suggestive. He had also discerned and exploited the mathematical simplicity to which the adoption of the concept of similarity leads. For this he is to be congratulated. It is our opinion that Professor Koestel's equations are more likely to be applicable to long-throw than to short-throw jets. For short-throw jets we are of the opinion that consideration must be given to the terminal zone where Professor Koestel's equations do not apply.

W. O. HUEBNER, New York, N. Y. (WRITTEN): In his paper Air Velocities in Ventilating Jets* presented at the annual meeting at Chicago in 1953, Professor Tuve gave a very complete summary of present day studies on axial and radial jets. This information is now incorporated in the 1954 GUIDE in the chapter on Air Distribution.

Some equations given by Professor Tuve in the paper were used by G. E. McElroy to prepare two charts for the chapter, namely: (1) a chart for determining centerline velocities of axial and radial jets, and (2) a chart of entrainment ratios and straight flow outlets. These charts may be easily applied for the solution of practical air distribution problems.

However, all this information deals with non-isothermal jets only. Professor Koestel's paper, therefore, by its very title, excites the interest of the reader.

(1) I am not in a position to judge if Professor Koestel's simplifying assumptions are correct. The extremely low value of the parameter $D_o \Delta t_n \beta g / V_o^2$ however, seems indeed to suggest that—as already pointed out in THE GUIDE—temperature and density differences have but small effect on cross-sectional velocity profiles of straight flow outlets.

(2) I can also offer new figures to either affirm or contradict Equations 3 and 6. However the equations check very well with the data of other investigators, and there seems to be no reason to assume that they are not valid for practical engineering purposes. It is particularly gratifying that Equation 9 so closely approximates Professor Helander's correlation equation.

(3) To increase the value of Equations 3 and 6 for the practical engineer, they should be translated into charts. There arises, of course, the question whether these charts would differ greatly from the charts prepared by Mr. McElroy, the second element of Equation 3 being so small.

(4) In addition to its main purpose of providing a basis for exploring the performance of non-isothermal jets, the paper adds to our general knowledge of jets. The characteristics of isothermal and non-isothermal have, to my knowledge, never been as clearly shown as in Figs. 2 and 3 of the paper. Also the whole method of analysis, though we may not be able to follow it in all its details, increases our understanding of the nature of ventilating jets.

(5) Prof. T. F. Hatch of the University of Pittsburgh Graduate School of Public Health is studying the basic principles of local exhaust ventilation for hot industrial processes by investigating convection currents set up by hot bodies. In spite of the difficulties mentioned in Professor Hatch's Progress Report of December 1953, Professor Koestel's equations might be of use in the former's investigation.

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* See Reference 1 of the paper.

R. C. EDWARDS, Pompton Plains, N. J.: In Fig. 1 of the paper a graph of the distribution of temperature and velocity in the air stream is shown. The graph indicates that the heat of the air stream is dissipated faster than the velocity. Since the conductivity of air is very small, why should this phenomenon occur?

AUTHOR'S CLOSURE: Thanks to Professor Helander and the other discussors for their excellent comments on the paper. I am sure that the limitations to the equations mentioned in Professor Helander's discussion and which are based on very careful observations will form an excellent contribution to the paper.

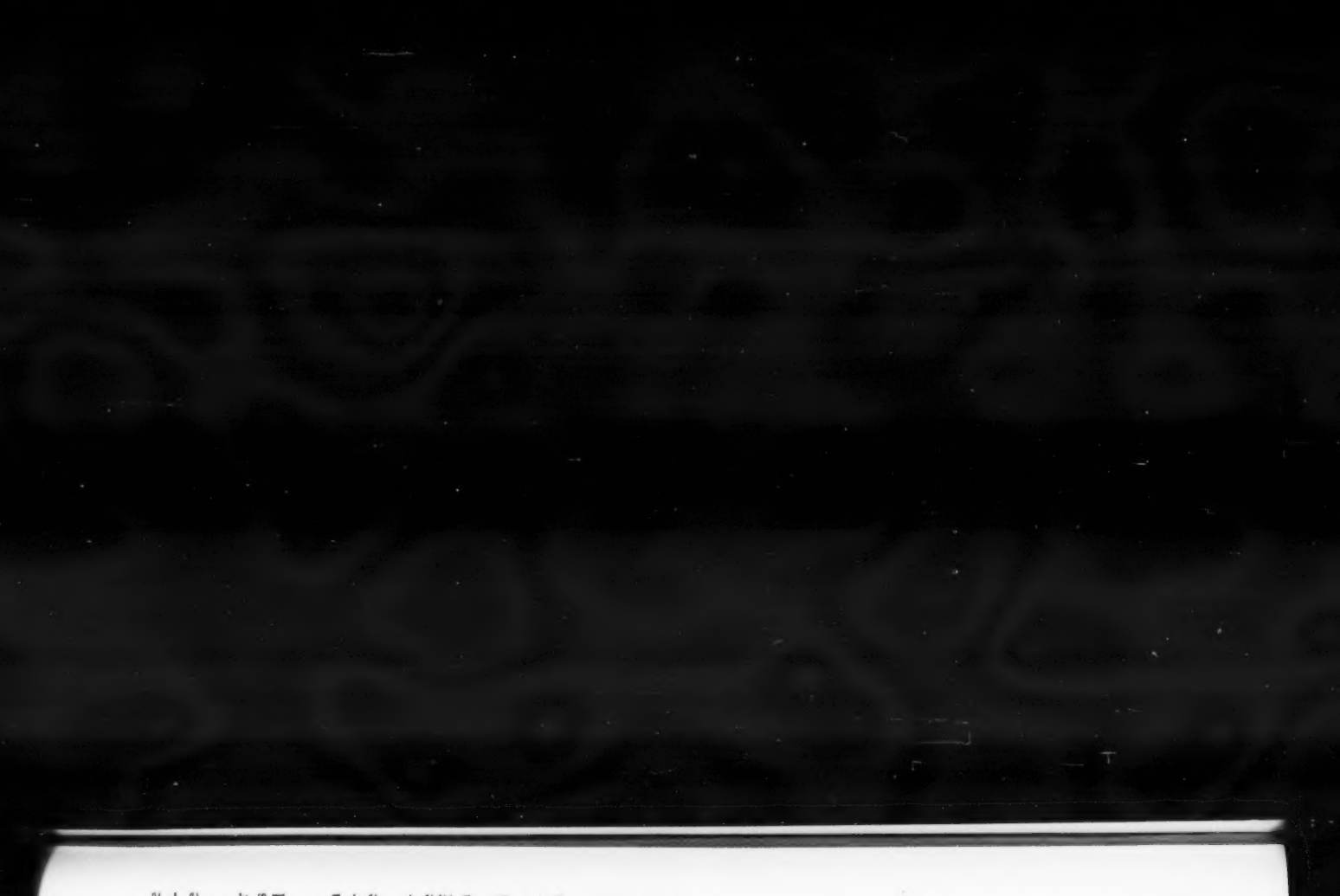
I question one point in his discussion and quote the passage as follows:

"For long-throw jets wherein the ratio of the buoyancy force to the momentum force is small, we agree that the existence of similarity may be an acceptable, simplifying assumption. We are skeptical of the acceptability of the assumption where the ratio of the buoyancy force to the momentum force is not small."

Fig. 9 in the paper contains data for a vertical jet of heated air in natural convection. It would be expected that the ratio of buoyancy force to momentum force would be exceptionally large for a jet of this type. Note also that the product of the temperature difference, the velocity, and the distance squared is a constant which would indicate that similarity exists since this product is based on similarity.

In reply to Mr. Huebner, I doubt if the charts in THE GUIDE would apply for all cases of vertical non-isothermal jets especially near the terminal regions of the jet. However, the isothermal jet equations may provide reasonable estimates for horizontally projected heated and chilled jets and for vertical non-isothermal jets in regions where the velocities are still relatively high.

In reply to Mr. Edwards question as to why heat is dissipated at a greater rate than momentum I would like to say that momentum (velocity) is dissipated primarily by turbulent mixing and very little by viscous action whereas heat (temperature) is dissipated by both turbulent mixing and conduction. Why heat and momentum do not diffuse equally is due to the physical properties of the fluid that we are concerned with.



1513

No. 1513

SOUND-PRESSURE LEVELS AND FREQUENCIES PRODUCED BY FLOW OF WATER THROUGH PIPE AND FITTINGS

By W. L. ROGERS*, EVANSTON, ILL.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the Northwestern University, Evanston, Ill.

THE INITIAL report in 1953 concerning this project outlined the experimental apparatus and methods of measurement being used[1]. In the present report, the first series of quantitative measurements and results are presented. The primary objective is to compare various pipe lengths and pipe fittings as noise sources at various velocities of water flow. Representative noise amplitudes and frequencies have been determined by means of a microphone mounted through the pipe wall and in contact with the flowing water. Measurements reported in this paper have been restricted to 1-in. pipe size and cold water of 37 F to 60 F. A limited amount of testing has been done at higher temperatures and also with larger size pipe, but these data will not be reported until a more complete understanding of the effects of these variables has been developed.

In the consideration of any noise problem, items of primary concern are the sources and the transmission of noise. It was desirable to investigate the noise sources first, and this report represents the first step in this study. The sound-pressure levels presented are those existing *inside* the pipe, and represent *potential* air-borne sound and structural vibration. The factors affecting the coupling of the vibrations which constitute sound in the liquid to the pipe wall and to the air are not a part of this report.

EXPERIMENTAL SET-UP AND INSTRUMENTATION

A tank 4 ft in diameter and 8 ft high serves as a water-supply reservoir. Steam may be admitted to the water in the tank to produce desired temperatures. In operation, the tank is not completely filled as air under pressure is admitted to the space above the water to produce desired pressures and flow rates. The water leaving the tank flows through 3-in. and 1½-in. piping to the test region. Rubber hose connects the supply pipe to the test length of pipe, which is sup-

* Assistant Professor of Mechanical Engineering, Northwestern University.

[1] A.S.H.V.E. RESEARCH REPORT No. 1481—Experimental Approaches to the Study of Noise and Noise Transmission in Piping Systems, by W. L. Rogers (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 347).

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1954.

ported by rubber loops spaced at random. The test pipe discharges into rubber hose, through an experimental acoustic absorber, through a globe control valve, and finally through additional rubber hose and piping to drain.

Operation involves a batch process, with control of pressures and flow rates in the test section achieved by controlling the air pressure above the water in the supply tank and the setting of the discharge globe control valve. No pumps are used, as they would add objectionable noise to the system. Flow rate is measured by timing the rate of drop of water level in the gage glass on the

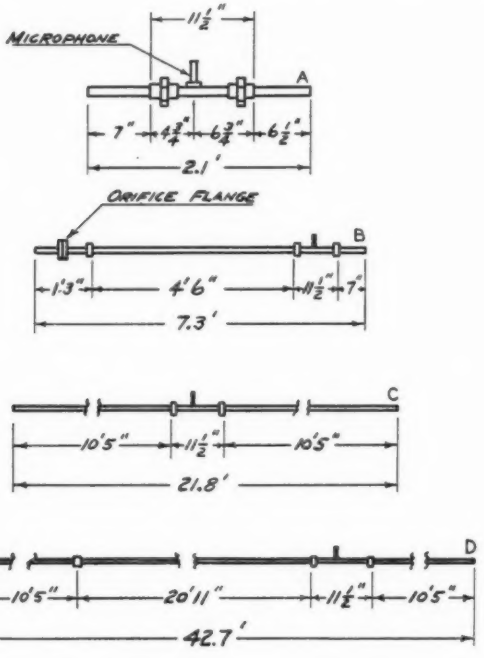


FIG. 1. MICROPHONE LOCATIONS IN THE FOUR TEST LENGTHS OF STANDARD 1-INCH STRAIGHT PIPE

supply tank. Pressure gages and thermometers are suitably placed for measurment of pressures and temperatures.

The instrumentation used in these tests for noise measurements consists of a microphone having a flat circular metal diaphragm ⅝-in. in diameter, plus associated electronic apparatus. The section of pipe into which the microphone is mounted is flattened slightly so that the microphone face is flush with the inside of the pipe wall. Before mounting the microphone, a coating of silicone oil is applied to the face, a thin rubber membrane is stretched across it and then the assembly is placed and held in the mounting hole. The membrane protects

the microphone from the flowing water, and the oil is used to insure good contact of the rubber with the microphone face, without air pockets.

Connected to the microphone are suitable amplifiers, an electronic voltmeter, and a frequency analyzer. Because the noises generated by flow have been found, in general, to be broad-band in character, it was decided that an octave-band analyzer should be used. This analyzer divides the frequency spectrum of 20 to 10,000 cycles per sec into 8 bands of frequencies as shown on the accompanying diagrams, and measures the level of each band. Adequate frequency information can be obtained with this type of analyzer in a relatively short time. There *have* been a few instances in which definite notes or tones have been heard through the headphones used to monitor the noise while recording data, but such instances have been rare to date.

TEST SET-UPS USED IN THE CURRENT TESTS

Fig. 1 in sections A, B, C and D shows the microphone locations and the lengths of 1-in. standard pipe used in the series of runs on straight pipe. The supply line to the left of the test pipe was soft rubber hose, and the discharge to the right consisted of soft rubber hose, an acoustic absorber, and a globe valve for controlling the flow rate, as previously mentioned. In spite of the considerable length of discharge rubber hose, it was found that additional attenuation of noise created by the control valve was required. It was therefore necessary to add an absorber. It is expected that future reports will deal with the evaluation of this type of experimental absorber and other control procedures.

At pipe-to-pipe and pipe-to-hose connections, precautions were taken to avoid appreciable and sudden changes in area in order to minimize unwanted background flow noise.

In comparing valves and fittings, it was necessary to hold pipe length constant so that this variable was eliminated. Arrangement B, Fig. 1 was used for this series of tests, with the fitting under test occupying the position of the orifice flange shown.

Practically all of the results presented were obtained with a pressure of 25 psig at the inlet to the test section. Occasionally runs were made with this pressure at 10 psig or some other pressure, and no significant differences in results were noted. A few of these points have been marked on the graphs. Runs were made with the microphone both upstream and downstream from the test components and, again, no significant differences were observed.

The primary variable against which noise levels have been plotted in this series of tests is velocity in standard 1-in. pipe, with pressure at inlet to the test section held constant (or having little effect), and temperature held to a relatively narrow range (37 F to 60 F). Flow was turbulent in all runs. During each run, sound-pressure level and octave-band frequency analysis were determined.

DECIBEL UNIT OF SOUND-PRESSURE LEVEL

The decibel unit used in this report to indicate sound-pressure level is defined in *American Standard Acoustical Terminology* Z24.1 - 1951 as

$$\text{db} = 20 \log_{10} (p/p_o)$$

where p is the actual sound pressure of interest and p_o is an arbitrary reference value. Two values of p_o are in common use. One is (2) (10^{-4}) microbar,

which is in general use for measurements made in air and for some measurements made in liquids. The other is 1 microbar, which has gained wide usage for calibrations and for many types of sound-level measurements in liquids. The latter reference level has been used here so that, specifically, the decibel unit used is defined as

$$db = 20 \log_{10} p$$

where p must be measured in microbars (one microbar is one dyne per square centimeter, or about one-millionth of an atmosphere). To convert from a decibel based on one reference level to a decibel based on the other is a simple matter.

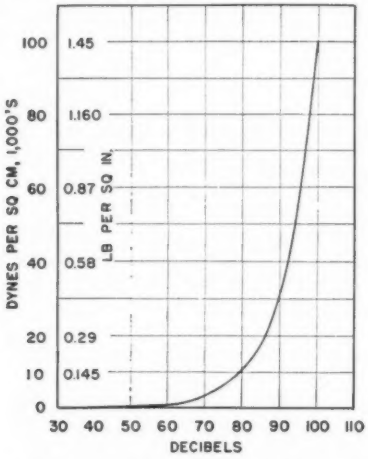


FIG. 2. SOUND-PRESSURE LEVELS IN DECIBELS REFERRED TO 1 DYNE/CM^2, AND CORRESPONDING SOUND PRESSURES

It is only necessary to add 74 to the db levels given in this report, and the result will be the db based on a reference level of (2) (10^{-4}) microbar.

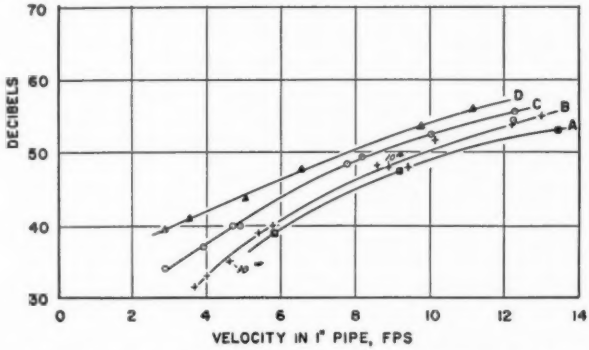
The use of a logarithmic unit in acoustic work is advantageous because of the great range of pressures ordinarily encountered, and also because it has been found that 1 db represents, roughly, the minimum change in an existing sound level that will be noticeable to average people. Fig. 2 includes the approximate decibel range covered in this report together with the corresponding sound pressures expressed in microbars (dynes per sq cm), and in psi. The db range from 30 to 90 involves a pressure range from 0.00046 psi to 0.46 psi, or a pressure ratio of 1000 to 1. Equal *changes* in decibels represent equal *ratios* of sound pressures. It is evident that decibel values cannot be added arithmetically.

NOISE PRODUCED BY FLOW OF WATER IN STRAIGHT PIPE

Fig. 3 gives the measured sound-pressure levels as functions of velocity of water in 1-in. pipe for the four different straight lengths of pipe, A, B, C, and D indicated in Fig. 1. The curves show that noise is increased as velocity is

increased and also as pipe length is increased. The curves tend to flatten at the higher velocities, indicating that velocity changes affect the sound pressure levels less at high velocities than at low ones.

Fig. 4 was plotted from the smoothed curves of Fig. 3. One trend indicated by this derived set of curves is that an increase in length of a pipe already long has less effect in producing additional noise than the same length increase to a short length. Also, it appears that if the velocity of water in the pipe is high, the noise level produced is not so sensitive to changes in pipe length as if velocity is low. This is particularly true for the shorter pipe lengths. The curves indicate that for very long pipes, the noise produced tends to become independent of length at any constant velocity.



The symbols A, B, C, and D refer to the test lengths indicated in Fig. 1. Two check points are shown for which inlet pressure was 10 psig. For all other points, inlet pressure was 25 psig

FIG. 3. SOUND-PRESSURE LEVELS PRODUCED BY FLOW OF WATER AT VARIOUS VELOCITIES IN STRAIGHT PIPE

Fig. 5 shows representative frequency spectra obtained by octave-band analyses of noise produced by flow in the three straight-pipe arrangements B, C, and D of Fig. 1. For the upper three curves, flow velocities were 10.15, 10.05, and 9.8 fps; for the lower three, velocities were 5.4, 4.9, and 5.05 fps. Probably the important items to be noticed are the general similarity of the frequency patterns of all three lengths and the fact that the predominant frequencies which largely make up the total level lie in the low-frequency bands.

These sets of curves for straight pipe represent patterns with which noises produced by various fittings will be compared in the following sections of this report.

It was necessary to eliminate the variable of pipe length when testing the various fittings and, for convenience, the basic test set-up shown in diagram B in Fig. 1 was employed for all of the following tests. The fitting under test occupied the position of the orifice flange.

NOISE PRODUCED BY FLOW THROUGH ORIFICES

In Fig. 6 are plotted the results of tests on four circular orifices. The largest had an area equal to the inside area of 1-in. pipe, and therefore the lowest

curve of Fig. 6 is merely a straight-pipe run with no orifice at all. It corresponds to the run marked B in Fig. 3. The other three orifices had flow areas of ½, ¼, and ⅛ of the cross-sectional area of the 1-in. pipe, and the curves of Fig. 6 are marked accordingly.

It is immediately apparent that orifices are potent sound sources, and that the sound-pressure levels produced can be extremely high. The two upper curves of Fig. 6 exhibit sharp breaks at particular velocities, with abrupt changes in noise level. At the breaks, there occurred very sudden intense audible (air-borne) hissing which was not present at lower velocities. Simultaneously, pipe-wall vibration near the orifice suddenly increased, as might be expected. It is believed that these *critical* velocities (for the pressure and temperature conditions present) marked the onset of intense and rather large-scale cavitation.

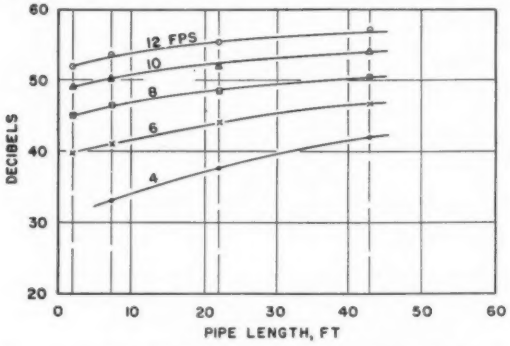
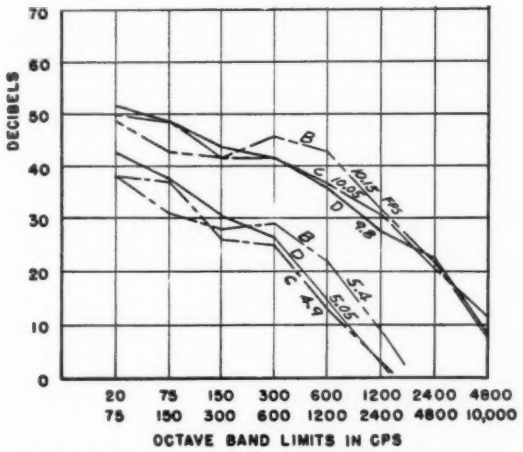


FIG. 4. THE EFFECT OF PIPE LENGTH UPON TOTAL NOISE GENERATED BY FLOW IN 1-INCH STRAIGHT PIPE AT VARIOUS VELOCITIES

Presumably the curve for the ½-area orifice (A/2 curve) would have exhibited the same characteristic break as the ¼ and ⅛ area orifices if the test had been carried to sufficiently high velocities.

The parts of the curves which lie on the left of the sharp breaks indicate that very high noise levels can be produced in the flowing water even without the presence of intense audible (air-borne) noise. A relatively small amount of audible (air-borne) noise *was* present at some of these lower flow-rates (although easily masked by other noises), but this noise did not approach the magnitude which appeared suddenly as the *critical* velocities were reached. Fast flow of water through an orifice unquestionably produces intense turbulence, with resulting large local variations of velocity and pressure. As flow rate increases, pressure must drop more rapidly through the orifice with resulting higher localized velocities and lower localized pressures. In other words, the localized extreme values of velocity and pressure cover a wider range, and the measured sound-pressure level reflects the greater magnitude of pressure variations. Eventually, a sufficiently low pressure can exist so that rapid release of vapor (or gas and vapor) bubbles, and the collapse of these bubbles as they move into regions of lower velocity and higher pressure, can cause a very sudden rise in the magnitudes of the pressure fluctuations constituting sound.

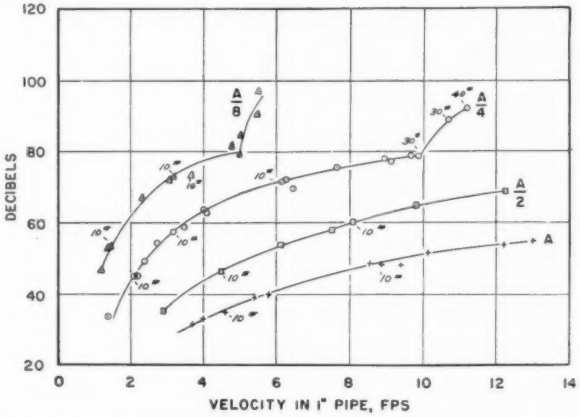


The symbols B, C, and D refer to the test lengths indicated in Fig. 1, and the numbers along the curves are flow velocities in fps

FIG. 5. REPRESENTATIVE FREQUENCIES PRODUCED BY FLOW IN STRAIGHT PIPE

These remarks at least present a reasonable explanation which is compatible with the experimental observations.

Although the *standard* inlet pressure used for all the tests reported was 25 psig, it will be noted in Fig. 6 that several check runs were made at 10 psig,



Inlet pressure was 25 psig except in those runs for which the pressure is marked 10, 30, and 40. The symbol "A" represents the cross-sectional flow area of the pipe test set-up B with no orifice

FIG. 6. SOUND-PRESSURE LEVELS PRODUCED BY FLOW THROUGH ORIFICES OF DIFFERENT SIZES

with negligible differences in resulting db values. For the ¼ area orifice, it was necessary to use higher inlet pressures (30 psig and 40 psig) to attain sufficient velocity to produce the sharp break in the curve attributed to cavitation. At these pressures the data also fit the pattern established by the 25 psig runs. It should not be implied that pressure is a variable which can be neglected. For cold-water tests, the pressure used apparently makes little difference because the water is far removed from a boiling condition. For hot water, it might be anticipated that pressure would become a more significant variable.

Fig. 7, plotted from data derived from the curves of Fig. 6, allows interpolation for orifice sizes between those that were tested experimentally. It is probably

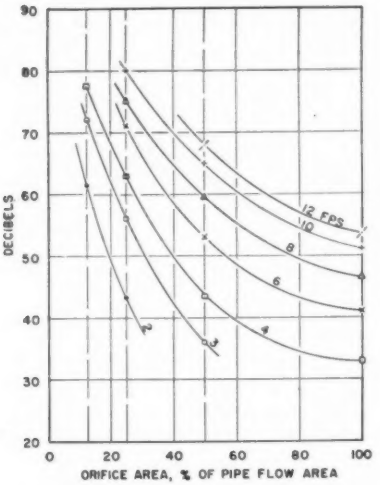
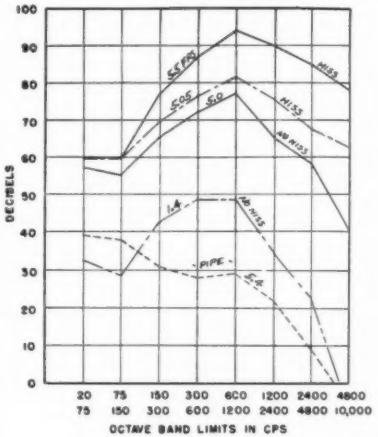


FIG. 7. THE EFFECT OF ORIFICE AREA UPON NOISE GENERATION FOR VARIOUS VELOCITIES OF APPROACH IN 1-INCH PIPE



Numbers on the curves represent velocities in the approach pipe in fps. The lowest short-dashed curve is for straight pipe only, without orifice

FIG. 8. FREQUENCIES PRODUCED BY FLOW THROUGH AN ORIFICE HAVING ⅛ OF THE FLOW AREA OF THE PIPE

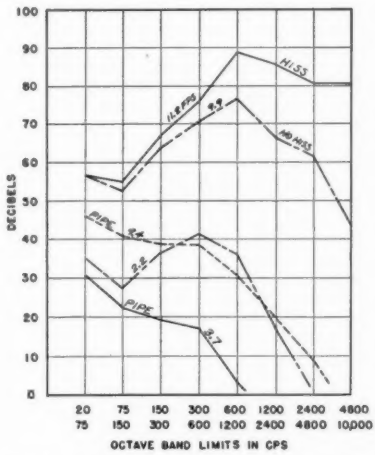
reasonable to expect that the general patterns indicated might be duplicated in pipe sizes other than the 1-in. size tested, but this assumption has not yet been checked. It should be noted that Fig. 7 utilizes only the data from Fig. 6 which lie to the *left* of the sharp breaks in the curves.

Figs. 8 and 9 present octave-band frequency analysis of noises produced by the ⅛ area and ¼ area orifices, respectively. In Fig. 8, the top curve represents the highest point on the A/8 curve of Fig. 6. The predominance of relatively high frequencies should be noted. The second and third curves from the top are for points immediately after and before the sharp break of Fig. 6. The velocities are very little different (5.05 fps and 5.0 fps), but intense audible hiss had developed at the higher velocity which was not present for the lower one. The higher-frequency levels, especially those in the 4800-10,000 cps band, increased proportionately more than those in the lower frequency ranges when

the hiss developed. For comparison purposes, the lowest curve of Fig. 8 (viewed in the 300-600 band) is plotted for a straight-pipe run with flow velocity of 5.4 fps. This velocity corresponds roughly with the velocities of the three uppermost curves already discussed. It is apparent that the straight-pipe noise is negligible by comparison with the orifice noise. The remaining curve of Fig. 8 is for a low velocity run of 1.4 fps, with the ⅛ area orifice in place.

It is apparent that as velocity through an orifice is increased, the total noise produced contains greater percentages of the higher frequencies. This seems reasonable. At higher rates of pressure drop across an orifice, it might be expected that pressure variations would occur at a more rapid rate.

Fig. 9, for the ¼ area orifice, shows the same general trends exhibited in Fig. 8. The top curve represents the highest point on the ¼ area curve of Fig. 6.



Numbers on the curves are velocities in the approach pipe in fps. The curves marked "pipe" are for straight pipe only, without orifice

FIG. 9. FREQUENCIES PRODUCED BY FLOW THROUGH AN ORIFICE HAVING ¼ OF THE FLOW AREA OF THE PIPE

Intense audible hissing was present. The second curve from the top, at 9.9 fps, represents the point immediately before the sudden break upward (Fig. 6). For this run, and also for that marked 2.2 fps (Fig. 9), audible hissing was absent. To be noted again is the rise in importance of the higher frequencies. For comparison purposes, runs at two different velocities in straight pipe without the orifice have been plotted. It is evident that the straight-pipe noise is of relatively little importance.

NOISE PRODUCED BY FLOW THROUGH VALVES AND ELBOWS

Fig. 10 presents measured sound-pressure levels for flow through elbows, wide-open globe and gate valves, and globe valves controlling flow. These items will be discussed separately.

Elbows: Four elbows were arranged to form a U-shape with two-foot sides and bottom. This arrangement replaced the orifice flange shown in Fig. 1 (diagram B), resulting in an increase in the total pipe length of the test set-up from 7.3 to 13.3 ft. Flow was started and the measured noise levels resulting are depicted in Fig. 10. The lowest curve in Fig. 10 is for the standard test length of 7.3 ft.

It is evident that flow through elbows produces more noise than flow through considerable lengths of straight pipe, especially at higher velocities. A comparison of the elbow curve of Fig. 10 with the straight-pipe curves of Figs. 3 and 4 substantiates this observation. For example, at 6 fps the elbow arrangement produced approximately 50 db. In Fig. 4, the curve of constant velocity

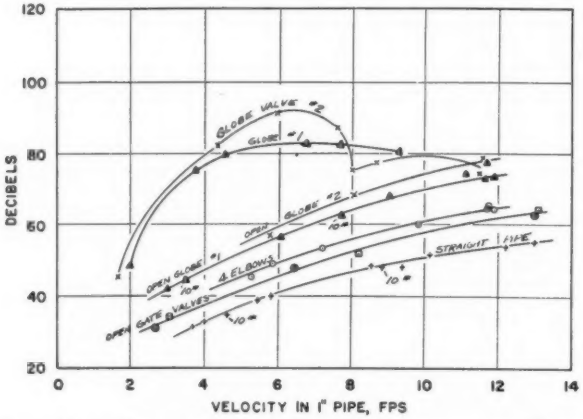


FIG. 10. SOUND-PRESSURE LEVELS PRODUCED BY FLOW THROUGH GLOBE VALVES CONTROLLING FLOW, GLOBE AND GATE VALVES WIDE OPEN, AND FOUR ELBOWS AND STRAIGHT PIPE

of 6 fps shows that 42.7 ft of straight pipe produced only about 47 db, and that additional length is less effective in producing additional noise.

The relative contribution of each of the four elbows to the total noise is not known. The problem of how to add properly the noises produced by multiple sources to determine a total noise level has been left to future consideration.

Fig. 11 shows representative frequencies for two velocities in elbows and, for comparison, two roughly corresponding velocities in straight pipe of length 7.3 ft (diagram B of Fig. 1). The frequency distributions are all quite similar.

Wide-Open Gate and Globe Valves: Fig. 10 shows the results of runs with wide-open settings of two samples each of gate and globe valves, selected at random. The gate valves were quieter than the four elbows tested and the points for both valves fall on the same curve. The globe valves were much noisier, and the two samples did not behave as nearly alike as did the two gate valve samples.

Frequency information is given in Fig. 12. Reading along the 20-75 line

and from the top down, the top two curves are for the two globe valves at the relatively high velocities of 9.05 and 8.05 fps. The next two are for a lower velocity of 6.05 and 5.75 fps. The following two are for the gate valves, at velocities of 8.2 and 6.45 fps. The bottom curve is a basic straight-pipe run at a velocity of 5.4 fps. There is a tendency for the higher frequencies to be of more importance in the globe-valve runs than in the gate-valve runs. As with orifices, the greater pressure drop required across the globe to produce a given flow rate probably results in more rapid pressure fluctuations.

Globe Valves Controlling Flow: The top two curves in Fig. 10 show that the two valve samples tested produced quite different noise patterns. The curve for valve No. 2 exhibits a sharp break and there appeared sudden intense audible

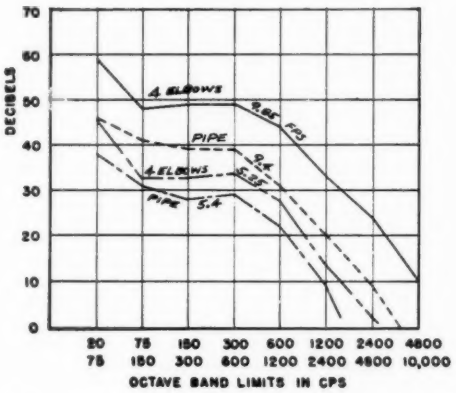


FIG. 11. FREQUENCIES PRODUCED BY FLOW THROUGH FOUR ELBOWS ARE COMPARED AT TWO VELOCITIES WITH THOSE PRODUCED BY FLOW IN STRAIGHT PIPE

hissing, along with the abrupt increase in decibels shown, as the valve was closed to throttle the flow. The characteristics observed were much like those of the orifices already discussed, including the maximum decibel values observed. The other valve (No. 1) developed an audible hiss gradually as the valve was closed, and did not exhibit the sudden and abrupt change of valve No. 2.

No detailed explanation of these differences in behavior will be attempted here. It is evident that the flow patterns inside the valves must have been somewhat different. Possibly one valve (probably No. 1) produced the total pressure drop with less abruptness than the other. The valves were not made by the same manufacturer, and valve No. 1 appeared to have been refaced many more times than No. 2. The fit of parts also seemed to be somewhat looser in No. 1.

Fig. 10 indicates that the magnitude of sound-pressure level produced by globe valves controlling flow reaches a maximum (with constant inlet pressure), and this can easily be observed by listening as a valve is gradually closed from a wide open position. At flow rates higher than those at the maximum sound-

pressure level, there is less pressure drop produced by the valve but a greater quantity of water involved. At flow rates below those at the maximum, there is greater pressure drop through the valve but a smaller quantity of water involved. These two items—the magnitude of pressure drop, and the quantity of water involved in the resulting flow pattern—need further study.

The frequency characteristics of globe valve No. 2 are shown in Fig. 13 along with a straight-pipe run at a velocity of 8.55 fps (lowest curve). For the valve curve at 8.0 fps, there was no audible hiss, while with the valve closed slightly to reduce the velocity, the hiss began. The hiss was steady at a velocity of 7.6 fps, reached a maximum of 5.95 fps, and decreased as the valve was throttled further to produce lower velocities. As with the orifices, a rise in importance of the higher frequencies occurred as audible hiss developed.

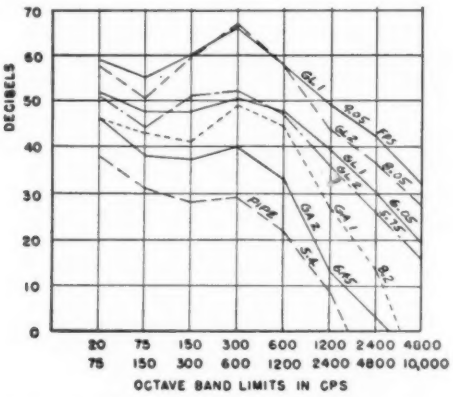


FIG. 12. FREQUENCIES PRODUCED BY FLOW THROUGH WIDE-OPEN GLOBE AND GATE VALVES AT VARIOUS VELOCITIES. THE LOWEST CURVE IS FOR STRAIGHT PIPE ONLY

(GL = globe valve; GA = gate valve)

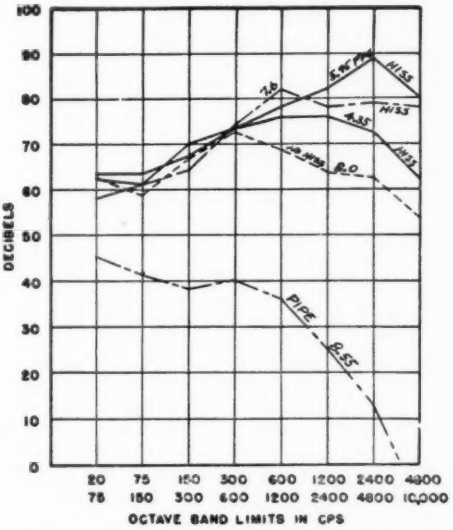
For valve No. 1, the distribution of frequencies was somewhat similar to that plotted for No. 2 in Fig. 13. However, as hisssing began, the higher frequencies were gradually apparent without the abruptness of valve No. 2.

Occasionally during the experimentation it was possible to find a flow rate for a flow-controlling valve which suddenly produced severe vibration of the valve, with accompanying definite tones and overtones. These could be heard by an observer in the test room, as well as through the headphones used for monitoring the microphone pickup. Such occurrences were relatively rare, and very slight changes in valve setting caused the phenomena to disappear. A possible explanation is the presence of a self-excited vibration because of looseness in the valve parts, in conjunction, perhaps, with a pipe of proper length to cause augmentation of the particular frequencies produced. With these few exceptions, flow noises were found not to consist of discreet frequencies, but to be rather broad-band in character.

CONCLUSION

Characteristic patterns and quantitative measurements have been determined for sound-pressure levels and frequencies produced in the water inside pipes by flow through representative piping components. The measurements indicate that elbows, valves, and orifices are all much more potent noise sources than straight runs of pipe.

The values presented are averaged readings, with fluctuations within the time intervals of reading sometimes amounting to several decibels. The plotted curves were spot checked occasionally, with reproducibility usually within 1



Velocities in the approach pipe appear along the curves. The lowest curve is for straight pipe only

FIG. 13. FREQUENCIES PRODUCED BY FLOW THROUGH GLOBE VALVE NO. 2 USED TO CONTROL THE FLOW

or 2 db. The tests are for cold water in the temperature range 37 F to 60 F, with 25 psig inlet pressure to the component under test (unless otherwise noted on the curves), and for 1-in. standard pipe size only.

Work in progress will extend the information to cover other pipe sizes and higher temperatures over the same flow velocity range. It is hoped that the data for different pipe sizes and different temperatures can be correlated in such a manner that results can be presented in compact form. Noises produced by pumps are to be evaluated, also.

After a sufficient background of information on noise sources has been developed, the problems of how to add the effects of these sources to determine total noise levels, the transmission of these noises to the pipe wall and to the air,

possibilities of reducing the magnitudes of the sources, and evaluation of methods of attenuation will be studied. Many of these items have already necessarily received some attention during the operation of this project, but they will be given more intensive study as the project continues.

ACKNOWLEDGMENT

The author acknowledges with thanks the continuing cooperation of Prof. B. H. Jennings, Chairman, Department of Mechanical Engineering, Northwestern University, and also the capable assistance of K. R. Johnson and W. D. Stoppkotte, graduate assistants.

DISCUSSION

H. A. LOCKHART, Morton Grove, Ill. (WRITTEN): Although there are an infinite number of tests that could be devised to establish the noise generating characteristics of fluid handling equipment, the more basic research of simple geometric shapes might be more fruitful than the investigation of specific prototypes of currently manufactured fittings. This does not mean that quantitative evaluation of pipe fittings now on the market is not needed—rather that is the stated objective of Professor Rogers' research effort.

Before tangling with the complexities of valves, it might be better to evaluate sudden enlargements, sudden contractions, gradual enlargements, gradual contractions, simple changes in direction, etc.

These suggestions are not meant as criticism of the excellent progress made so far, but rather as encouragement to go even farther.

R. T. KERN, Fitchburg, Mass. (WRITTEN): The experimenters on this project have done a great amount of work since they presented their first paper at Denver* last year.

However, it is my belief much of the work now being done will be of little value to the designing engineer unless the trend of the studies is changed.

I am under the impression this work was authorized as a study to ascertain the causes of noise in hot water heating systems and what steps are necessary to prevent, or at least lessen, those noises.

It was generally agreed, I believe, even before any tests were started, that probably the noise due to velocity in straight pipe was almost negligible and that the objectionable noises were probably caused by orifices and throttled valves used as controls.

Maybe it was necessary to use such high pressures and high velocities as shown in the figures presented in order to get at the basic facts. If this is necessary, then as little as may be necessary should be done and tests should be carried on at velocities normally used in heating systems in order to prove or disprove the theories deduced from the tests already made.

All the tests so far reported are for velocities of almost 3 fps to 13 fps and then with cold water varying from 37 to 60 F.

Heating systems use water in the vicinity of 175 to 200 F. There may or may not be a difference in the results between using cold and hot water but since the designing engineer is interested in hot water and not cold water, should not the tests, for the time being, be confined to the use of hot water?

The velocities used in the 1-in. pipe under test would approximate 600 milinches loss per ft for 3 ft velocity and 3000 milinches per ft for 10 ft velocity, all of which are far beyond the normal losses permissible in any but very special cases.

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* See Footnote 1 in the paper.

What is needed are answers as to what can be done when losses are between 150 and 600 milinches, the commonly used minimum and maximum allowable losses for 1-in. pipe, representing velocities of 18 in. to 36 in. per sec, and if there is no detectable noise at these velocities, what is the highest velocity permissible before noise becomes apparent.

Fig. 6 shows noise levels through orifices of various areas. The curves A/8, A/4 and A/2 would be for orifices having diameters of approximately 0.375-in., 0.4725-in. and 0.6675-in., respectively.

Table 4 on page 529 of THE GUIDE 1954 shows that such orifices, at the velocities reported, would be useless for normal design conditions; a 0.35-in. diameter orifice in 1-in. pipe having a pipe velocity of 1 fps would be 32,000 milinches; a 0.45-in. orifice at 2 fps would be 37,000 milinches while a 0.65-in. orifice at 3 fps would be 10,000 inches.

Using the same formula to carry the velocities beyond those published, as Dr. F. E. Giesecke used in preparing the table in THE GUIDE, the 0.375-in. orifice at 3 ft velocity would have a loss of approximately 91,500 milinches and the 0.4725-in. orifice at 5 ft velocity would be 82,000 milinches or 82 in.

Would it not be possible, for the present, to confine the experiments to the velocities and temperatures normally used and thereby get the information, so badly needed, in the hands of the designing engineer that much sooner and possibly at considerably less expense?

Following the study of fixed orifices it would seem, gate or globe valves should be studied in order to determine the greatest pressure drop permissible through them without creating undue noise and after that, other methods or designs of orifices which would accomplish the desired results without noise.

As one who has done a reasonable amount of designing large hot water heating systems, I think I speak from experience, and am sorry it seemed necessary to be so critical of a paper, which, on the face of it, shows so much thought and effort. However, I feel it will be a long time before usable information is forthcoming unless the trend of the present tests is changed.

H. C. HARDY, Chicago, Ill. (WRITTEN): The author is to be highly congratulated for studying a very important problem and presenting information to the Society. I am very pleased that he introduced considerable modern technology into the paper; for instance, the frequency measurements in octave bands. It is an oddity that the first publication with modern acoustic measurement techniques in the A.S.H.V.E. TRANSACTIONS should be on pipe noises rather than airborne noises with which members of the Society have worked so long, but it is understood that the latter technical information deficit will be reduced in the near future.

It should be emphasized that the data of Mr. Rogers' paper are not yet ready to be put into use in practical engineering problems. These data apply only to sound pressure levels in the fluid in the pipe. They should not be confused in any way with sound pressure levels in the air around the pipe, with which there will be, in general, no simple correlation. As Professor Rogers indicates in his conclusion, it is still necessary for us to learn how the amplitudes of vibration of the pipes are related to the interior sound pressure levels and how the amplitudes of vibration create sound in the air by direct coupling or by exciting through hangers the vibration of partition walls.

Previous work in our laboratories has shown conclusively that there is a strong coupling between the pipe vibrations and the interior sound pressure level. This coupling is so strong that a few feet from an excitation in the liquid, most of the energy is in the pipe itself. Might we suggest, therefore, that after the fundamental research relations are worked out by Professor Rogers and others, we depend principally on data determined by vibration measurements rather than on water coupled microphones. By this means the vibration data will be so much more conveniently

obtained in both the laboratory and the field and will, I believe, be less susceptible to engineering errors.

It is interesting to observe from Professor Rogers' data that the noise level rises approximately as the fifth power of the speed. However, it appears that one would have to exceed 50 fps in order to obtain speeds (below cavitation) which would make the flow noise important. This is also almost the same situation in air-flow systems.

May we hope that this pipe noise research will be continued with the same dispatch that has been done on this project thus far.

E. L. WEBER, Seattle, Wash. (WRITTEN): The entire paper, especially Fig. 6, is extremely interesting and educational. It clearly shows that objectionable noises are not due to the velocity of water through the pipes but are mainly due to the turbulence produced by orifices, fittings, valves, etc.

Fig. 10 clearly indicates that globe valves should never be used in water distribution systems. In these tests the noise produced by angle valves, in various stages of closure, should also be studied, as this type of valve is almost universally used as shut-offs for plumbing fixtures and for radiator valves, which occasionally produce undesirable noises when throttled.

The energy used in producing noise is added to the friction head to be overcome in the distribution system. Hence a solution of noise problems will automatically provide a solution to friction head problems. The two appear to go hand in hand.

Table 1, on page 525 of THE GUIDE 1954, shows a noticeable coincidence between the friction head through tees of various bores and the findings shown in Fig. 6 of the paper. Our firm prefers the use of all-welded pipe networks for hot water and panel heating systems and indicates branch take-offs from the mains by an arc connection similar to the branch take-offs customarily used in ventilating systems to reduce friction and noise (see Fig. 15, page 723, THE GUIDE 1954).

A study of various radii of pipe bends and of branch take-offs for minimum permissible noise level and friction should be of great value to the industry and could stimulate the manufacture of properly designed weld bends and branch take-offs, which could well compete in price with the hit and miss type produced in the field.

I am pleased to note that the tests are being conducted at velocities up to 13.5 fps and feel that velocities up to 20 fps in distribution mains will not be unusual in the future in high velocity water heating systems.

R. C. EDWARDS, Pompton Plains, N. J.: Does the author have any data on the conversion factor between the figures obtained and the noise level as observed by the ear, for example, at a distance of one or two feet from the pipe? If he has such a conversion factor, is the factor a constant?

Suppose the tests were conducted in a highly absorbent sound-proof room, having walls constructed of 6 or 7 in. of muslin covered rock wool. In such a room the ambient noise level could be as low as 15 to 20 db. Under such conditions wouldn't it be possible to take all readings at a distance of 1 ft from the pipe? Afterwards, a table could be worked out whereby, knowing the length of the pipe, the surface area and the fluid velocity in the pipe—the resultant noise level in a given space could be approximated. The use of such a table would, of course, presuppose that the pipe line would be well isolated by suitable hangers from the building.

I know from wide experience that such an approximation can be made in determining the noise level of air distribution devices.

You can make such a calculation with the distribution of air from outlets; therefore I feel that the same approach would apply with a piping system.

In summary, if you know the overall noise level of a piece of equipment at a certain distance and the absorption of the room, you could then, from a practical standpoint, make an approximation of the resultant noise level.

Therefore, it appears to me that if Mr. Hardy wants to obtain a practical answer,

work should be done in a highly absorbent sound room and some correlation charts be made between the fluid velocity in the pipes, the size of the pipe, and the absorption of the room.

JOHN W. JAMES, Chicago, Ill.: I am not an acoustical engineer but I am Chairman of the Technical Advisory Committee on Hot Water and Steam Heating, which is supervising the studies reported by this paper.

I have one question for Professor Rogers, one which deals with the interest that some people have in snow melting today where liquids or oils or glycols are being added to water. I was wondering whether he would care to comment on the effect of increased velocity, using such liquids, on some of the results he has obtained so far.

I am glad that Mr. Kern spoke as he did because we felt that sooner or later our committee would have to justify the continuance of this research project, which the Society is helping to support in various ways.

When we first tackled this problem a few years ago we thought the answer was easily obtainable on a basis comparable to the approach Mr. Kern presented to you this morning. We have become educated in the last few years meeting together as a group and talking with Professor Rogers, and have found that if we were to get the answer we would have to spend a little more time than we hopefully thought at the first, because what we were primarily interested in was the thing that Mr. Kern spoke to you about earlier. That is, we wanted the answer to problems in connection to sounds the normal person hears in hot water heating systems. Unfortunately, we couldn't get at it in any other way from our analysis of the problem than the way Professor Rogers has led us to this point.

We are now at the brink of what we feel is a very opportune time to actually get a lot of data which will be useful to the Society membership. I hope you will bear with us just a little longer.

I am reminded of the time a few years ago when the Society spent some exploratory money at Renssalaer Polytechnic Institute to study sound absorption in air ducts. Unfortunately, we didn't continue because of expensive and time consuming basic research work that was required; so the data we have today I think is largely wrapped up in the hands of the manufacturers. They have done a wonderful job in utilizing commercial sound absorption units in air distribution systems, but unfortunately we are not in the same position in water systems today.

Professor Rogers has shown us some of his ideas as to what might be done along these lines, but they are just ideas today. Those are the things that are going to come out of this study and we are very hopeful that if you will stay with us a little longer and support us financially, we are going to have some answers for you.

E. R. TESKE, Chicago, Ill.: In relation to the question as to whether noises in piping systems are ever a problem it has been our experience that any audible noise in living quarters or sleeping quarters at night is objectionable, and most frequently the heating system is blamed.

With regard to the temperatures and pressures that Mr. Kern suggested as being more practical, we feel that the temperature range selected by Mr. Rogers is a practical starting point. Any data of this type must eventually be applicable to low temperatures experienced in chilled water systems up through those found in high temperature water systems.

We have knowledge of noise difficulties being experienced with water circulation piping running through offices to serve a 200-ton cooling tower. At the opposite extreme of temperatures a building nearby is served by a 350 F hot water heating system.

We have found that frequently cavitation noise sources can be eliminated by raising the static pressure on the system. In a further study of noise produced by pumps, etc., we hope that the author will give consideration to a possible pipe organ effect. We

have experienced situations where noises could be heard at certain remote locations in the piping system and yet could not be heard at what was determined as the point of origin. We realize this is a very difficult problem because of the variations in piping installations, but suggest that at least some investigation be made as to the sound levels produced by such effects.

A. T. JONES, Toronto, Ont., Can.: In plotting Fig. 4, Professor Rogers used the overall length of the test pipe regardless of the location of the microphone. It would appear at first thought that the distance from the entrance of the test pipe to the microphone should have been used in plotting Fig. 4. Is there any explanation for using the whole length which in several cases extends a considerable distance beyond the microphone.

It would seem to me that additional turbulence and vibration may have been set up by the microphone itself and the flattening of the pipe where the microphone was installed. What tests were made to ascertain whether any vibration set up at this point was of unimportant magnitude?

Without attempting to answer these questions, I presume the primary object of the paper was to show differences and trends rather than quantitative information which is in this work difficult to obtain and difficult to visualize.

As Mr. Kern said, we are all very anxious to arrive at final definite information that can be given to the industry to assist in design work. Questions continually arise as to whether system piping should be designed on a maximum or 300, 400 or 500 milinches or on a maximum velocity of say, 2, 3 or 4 fps to avoid excessive sound or vibration due to water flow.

It seems quite possible that Professor Rogers will finally arrive at some sort of chart giving permissible flow conditions—it might almost be called a comfort chart. It will likely show that velocities that are permissible in larger pipes are excessive for smaller diameters.

I am sure we are all looking forward to seeing this work applied to a range of diameters and conditions and also to a comparison or relationship between sound and vibration inside piping to that transmitted to the building structure and to surrounding air.

W. M. WALLACE, II, Durham, N. C.: My question is rather in the form of a question and a suggestion, unless Professor Rogers can answer the question.

I am very much interested in knowing whether isolating a pump with sections of rubber tubing attached to the suction and discharge piping, will effectively reduce the noise of the pump being transmitted through the building when the pump is located in a boiler room or mechanical room. If Mr. Rogers is not in a position to make any definite observation on this, I would suggest that at some time in the near future, experiments be conducted to determine the effectiveness of rubber tubing connections in reducing the transmitted noise.

AUTHOR'S CLOSURE: We wish to thank the discussors for their comments, questions, and suggestions, all of which contribute additional experience and ideas, and help in formulating the proper direction of the project.

When starting this project, we did try to measure the sound level outside the pipe with a microphone, as suggested by Mr. Edwards. We learned that there was apparently no simple relationship between the noise generated in the water and the air-borne noise. The air-borne sound was much less intense, of course, but also the distribution of frequencies was different. The manner of supporting the pipe influences the natural frequencies of the piping arrangement, and therefore influences the types and frequencies of pipe motions which produce air-borne sound. We decided that we should adopt the more fundamental approach of trying to measure and rate the noise sources right in the water. This would allow evaluation of various fittings and

components as noise-producers, with the hope that some common denominator (such as abruptness of pressure change) could be found. The matter of how the sounds produced in the water by the sources are transmitted to the surroundings of the water is then a separate problem. Some work has been done on this problem by Calloway, Tyzzer, and Hardy (*Acoustical Society of America Journal,* Vol. 23), and by the Navy, so that some helpful information is already available. We hope to be able to add to this information eventually.

We might mention here that air-borne levels of noise were determined, essentially by the scheme of Mr. Edwards, by F. W. McGhan, and published in the *Plumbing and Heating Journal,* Jan. 1952. It should be realized, however, that noise (vibration) travels along the water path, along the metal of the pipe, and, partially, through the air as a result of the pipe-wall vibration. The determination of the division of vibration (noise) among these paths of transmission is not a simple matter. Knowledge of only one of them does not solve the potential problems which might be created by the others at some other location, where more favorable coupling circumstances may exist. In other words, the immediately air-borne sound is often not the main problem.

In studying the coupling of vibrations (sounds) in the water to the pipe, to the air, to building panels through supports, etc., it probably would be helpful to include a procedure such as that proposed by Mr. Edwards.

Mr. Hardy suggests the eventual use of pipe-wall vibration measurements to replace measurements using water-coupled microphones. If a satisfactory correlation could be determined, such a procedure could probably simplify the measurement problem.

We appreciate and share Mr. Kern's impatience to get practical answers as soon as possible, and believe that our viewpoints are not too far apart. We already have done some of the obvious things which he suggests, such as the use of hot water, but much of this information and data simply could not be readied for publication in time for this meeting. The paper presented should be considered as an *in progress* report.

As to basic philosophy regarding this investigation, it is our belief that a fundamental approach in which basic information and understanding are sought will, in the long run, yield the greatest benefit for the time and energy expended. Such an approach may not immediately solve many practical problems, but the information eventually forthcoming may shed light on a much broader range of situations and applications than would a study limited to only certain specific practical matters of the moment. That is the hope and expectation for this project. We expect to cover a range of velocity, temperature, and other variables which is broad enough to be useful for a long time. Also, a certain range is necessary if trends and limits are to be determined and established. At the moment, we must face the limitations of certain instrumentation, especially the temperature and sensitivity limitations of available microphones which will fit our space requirements. As for water temperature, the problem of noise in air conditioning systems arising from flow of cold water through various components can be as great a problem as noise in hot water heating systems and would appear to be a problem which is definitely within the scope and interest of the A.S.H.V.E.

Mr. Lockhart's basic viewpoint is appreciated. Understanding of the effects of simple shapes can add a great deal to understanding of what happens in more complex apparatus.

The question asked by Mr. James cannot be answered adequately at this time.

To answer Mr. Jones, it might be noted that sounds travel large distances in water with very little attenuation. Noise created by flow through any small section of pipe could be expected to be transmitted throughout the whole pipe test section with very little attenuation. In other words, each little section of pipe contributes its share of total noise present in the pipe. To sound waves travelling at several thousand feet

per second, the presence of some flow velocity has negligible effect, so that the terms upstream and downstream have little meaning when thinking of the travel of the sound waves. To test this idea, the pipes were turned end for end in several tests, with no change in the measured sound levels. If a noise consisting of a discrete frequency were present, so that resonance and standing waves might result, then microphone placement would become a more important item. For the tests reported, however, the noises were of broad-band character.

The microphone mounting arrangement, in our opinion, was responsible for only a minor amount of noise production. Whatever it was, however, it is included, along with the noise produced by flow through 2.1 ft of pipe, in the values given by curve A of Fig. 3. Now, when a new noise level is superimposed on an existing level to give a total reading which is 10 db or more higher than the previously existing value, the total reading is practically the level of the superimposed new sound, and the correction for background level is negligible. When the difference is less than 10 db, a sliding scale of corrections is in order. For most of the tests made, no correction was necessary. Therefore, the values presented are substantially correct quantitatively, as well as presenting trends and differences.

We are aware of the situations described by Mr. Teske and hope to include some resonance or *pipe-organ* tests at some future time. There are probably situations in which certain frequencies can be amplified by resonance conditions.

Mr. Weber's statement that "a solution of noise problems will automatically provide a solution to friction head problems" may be an over-simplication, but we agree that in many cases the two are related.

Mr. Wallace asks about the effectiveness of rubber tubing as a sound absorber. This was discussed in a general way in the first paper presented on this project, and is published in the A.S.H.V.E. TRANSACTIONS of 1953*. We have no quantitative information yet for publication, but it was necessary for us to have effective attenuation before and after our test section to block noises originating upstream and downstream from entering the test region.

Effective procedures were evolved after consideration experimentation. However, the subject of quantitative evaluation of means of reducing sound transmission will be reserved for later reports. At present, the focus is on the noise sources, although it has been necessary already to devote considerable attention to methods of attenuation.

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* See Footnote 1 in the paper.

1514

No. 1514

COMPUTATION CHARTS AND THEORY FOR RECTANGULAR AND CIRCULAR JETS

By HAROLD G. ELROD, JR.*, CLEVELAND, OHIO

A CONSIDERABLE body of experimental information is now available concerning the velocities in fluid jets having discharge openings of various shapes. For the designer of systems employing jets, it is very desirable that this information be correlated in a compact, accurate, and simple manner. Although considerable success has already been achieved in the making of such correlations, it is believed that the present correlation based on Reichardt's[1] free-jet theory is more satisfactory than any heretofore obtained. This correlation has been incorporated in the charts given in Figs. 1, 2 and 3. These charts are substantially independent of the accompanying test, so that the reader, if he desires, may apply them immediately for his own purposes.

The theoretical basis for the present work lies in the free turbulence equation of Reichardt (Equation 1). Although not as well founded in basic fluid-mechanical hypotheses as the formulations of Prandtl and Taylor, this equation gives rather better agreement with free-jet experiments and is, furthermore, much more susceptible to mathematical analysis. Being a linear equation, it yields results for finite sources and for simple multiple-jet systems by the method of superposition. Since it can correlate a wide variety of known jet phenomena by means of a single empirical constant, it may be used with reasonable confidence to interpolate or extrapolate to unstudied situations. However, it is known that Reichardt's equation is not applicable when static-pressure variations within the jet are appreciable.

Many experimenters have corroborated the following three observations concerning free jets:

1. The momentum of a free fluid jet is approximately conserved. That is, external pressure and viscous effects are generally negligible.
2. At distances z remote from the jet-source, the momentum flux (ρW^2) distribution as a function of displacement r from the jet axis is well approximated by the Gaussian error curve.
3. A free jet spreads with a nearly constant angle of divergence.

The foregoing observations lead uniquely to a momentum-flux distribution which is identical in mathematical form to temperature distribution resulting

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* Assistant Professor of Mechanical Engineering, Case Institute of Technology.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1954.

from the diffusion of heat from an instantaneous line source of energy in an infinite, conducting medium. In the first instance it is momentum, and in the second instance it is energy, which is conserved. Because of the similarity Reichardt hypothesized that *momentum diffuses with distance from a source in the same manner that heat-energy diffuses with the square-root of time.* This simple correspondence permits an attack of problems in jet propagation by use of accumulated experience in heat conduction. In particular, known solutions from this well-established field can be borrowed to provide directly the characteristics of a variety of fluid jets.

The differential equation for free turbulence follows naturally from Reichardt's hypothesis. Thus, by analogy with the well-known equation for heat conduction, Equation 1 may be written:

$$\frac{\partial \rho W^2}{\partial (z^2)} = \frac{c^2}{4}\left[\frac{\partial^2 \rho W^2}{\partial x^2} + \frac{\partial^2 \rho W^2}{\partial y^2}\right] \quad \ldots\ldots\ldots\ldots\ldots (1)$$

Symbols used in Equation 1 and following equations are defined under Nomenclature.

This differential equation forms the basis for all the theoretical results given in this paper. It also furnished the starting point for the recent work of Alexander[2] in which the theory was shown to reproduce satisfactorily the velocity field near the outlet of a circular nozzle.

COMPUTATION CHARTS AND THEIR LIMITATIONS

Fig. 1 shows the velocity profiles generated by a rounded-entrance, infinite slot. The reader may observe here in a semi-quantitative manner the dissolution of the original *platform* profile into the customary Error Curve. The series of

NOMENCLATURE

A = cross-sectional area of region through which jet is passing, square inches,† dimension L^2.
a = half-width of infinite slot, inches,† dimension L.
b = half-width of infinite slot, inches,† dimension L.
c = empirical constant in Reichardt's turbulence Equation 1.* Approximate value: 0.08.
C_d = discharge coefficient of jet aperture.*
D = diameter of circular aperture, inches,† dimension L.
E = entrainment ratio of jet.*
i = square root of -1*.
J_p = Bessel function of order p.*
o = subscript for source.*
p = exponent in series expansion.*
r = distance from axis of circular jet, inches,† dimension L.
t = auxiliary variable.*
W = time-mean velocity in z-direction, feet per minute,† dimensions L/T.
x = auxiliary variable.*
x = Cartesian coordinate in plane of jet source, inches,† dimension L.
y = Cartesian coordinate in plane of jet source, inches,† dimension L.
z = Cartesian coordinate in plane of jet source, inches,† dimension L.
ω = parameter in Bessel-function formula.*
ρ = fluid density, assumed constant, dimensions FT^2/L^4.

†Or other consistent units.
*Dimensionless.

profiles expresses the following relation, which must be used if considerable inaccuracies from interpolation are to be avoided.

$$\frac{W}{W_o}=\sqrt{\frac{\operatorname{erf}\left(\frac{a+x}{cz}\right)+\operatorname{erf}\left(\frac{a-x}{cz}\right)}{2}} \quad \ldots\ldots\ldots\ldots \quad (2)$$

In this equation W_o represents the peak velocity of the source. The function "erf (x)" is the Probability or Error Integral, and is tabulated in many places, *e.g.,* Reference 3.

Fig. 2 gives peak (or center) velocity ratios for jets from rounded rectangular sources of all aspect ratios. The term *velocity ratio* is reserved in this paper

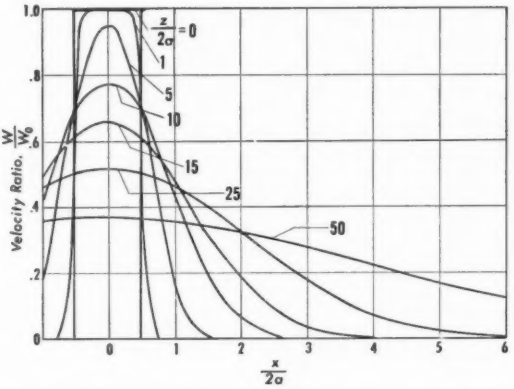


FIG. 1. VELOCITY PROFILES FOR THE ROUNDED INFINITE SLOT

exclusively to express the ratio of the local time-mean velocity in the axial direction to the original peak velocity of the jet source.

Fig. 3 presents entrainment ratios for jets from rounded rectangular sources. This entrainment ratio is a measure of the amount of fluid circulating in the axial direction within the jet. It is defined as the ratio of the volumetric flow of the jet across a plane some distance z from the source to the volumetric flow emitted by the source. (A fluid of constant density is assumed.)

These first three figures purport to describe in detail the characteristics of fluid jets over extended ranges of dimension and velocity. The correlation on which they are based, however, has its limitations. These limitations will now be outlined, as the author presently understands them.

Nature of Jet: The charts are for *free* jets of *any constant-density fluid* discharging in one direction into a stationary field of the *same fluid.* The density and viscosity of the fluid affect the correlation only if the Reynolds number condition given is not satisfactorily met.

Nature of Jet Surroundings: Obviously, no jet is *free* in the sense of having characteristics which are completely independent of its surroundings. The data of Nottage[4] and of McElroy[5] appear to indicate that a jet may be con-

sidered as free until the cross-sectional area of the jet equals about one-fifth of the total cross-sectional flow-area of the region through which it is proceeding. For those cases where the source area itself is small compared with the region area A, the foregoing limitation is that the jet may be considered free so long as

$$z \leq 1.5 \sqrt{A} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (3)$$

Source Reynolds Number: Becher[6] reports that a jet ceases to be laminar when the source Reynolds number (based on the hydraulic diameter of the jet outlet) reaches about two thousand. The present correlation, however, is based on *fully developed turbulence*. Because of contradictory evidence for source

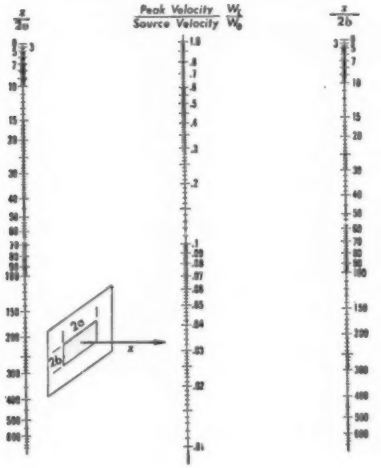


FIG. 2. NOMOGRAM FOR CENTER, OR PEAK, VELOCITIES FOR ROUNDED, RECTANGULAR JET SOURCES

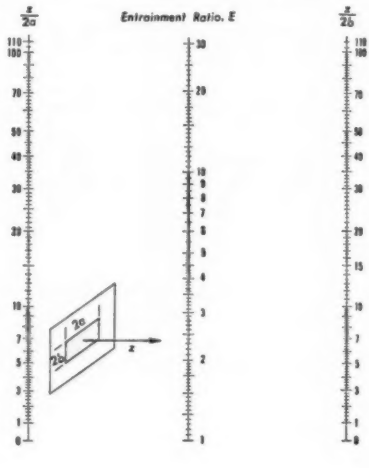


FIG. 3. NOMOGRAM FOR ENTRAINMENT RATIOS OF ROUNDED, RECTANGULAR JET SOURCES

Reynolds numbers less than 20,000, it seems best to choose this value as the lower limit of applicability of the computation charts. For Reynolds numbers lower than 20,000, the charts may predict velocities which are too high.

Jet Velocities: The source velocity must be low enough to prevent cavitation or compressibility effects. If a Mach number of 0.35 be set as the upper limit of *incompressible* flow, then air velocities under standard atmospheric conditions should be less than 20,000 fpm. Also, the accuracy of low velocity predictions can be upset by convection currents. Thus air-velocity predictions of less than 75 fpm must be considered as unreliable.

USE OF THE COMPUTATION CHARTS

The use of the charts will now be illustrated by specific numerical examples. Two salient jet features, the center velocity and the entrainment, can be readily

computed by the use of Figs. 2 and 3. For velocities other than those on the jet axis, only order-of-magnitude results can be obtained from Fig. 1. Frequently, such precision is all that may be needed. In general, if higher precision is desired, recourse must be had to formulas. Suitable formulas will be given in this section. For simpler expressions which are valid for restricted zones of a jet, the reader may refer to the next section of this paper, or to Reference 7, which covers the Error Curve Zone for a wider class of jets than is here discussed.

Infinite, Rounded-Entrance Slots

Given: A rounded, slot source, 1 in. $\leq x \leq$ 1 in. (a = 1 in.) with W_o = 3000 fpm.
Required: (a) The peak velocity 4 ft from the source.
(b) The entrainment ratio 4 ft from the source.
(c) The velocity at a point P with coordinates z = 48 in., x = 5 in. and y = 5 in.
Solution: (a) In Fig. 2 use $z/2a = 24$, $z/2b = 0$.
Then $W_c/W_o = 0.53$ and $W_c = 1590$ fpm.
(b) In Fig. 3 use $z/2a = 24$, $z/2b = 0$.
Then $E = 2.65$.
(c) From Fig. 1 can be obtained a rough estimate of the velocity at P.
Thus with $z/2a = 24$ and $x/2a = 2.5$, we read $W/W_o = 0.25$.
Therefore, W = 750 fpm. For a more accurate estimate, use Equation 2, remembering that $\operatorname{erf}(x) = -\operatorname{erf}(-x)$.
For the present problem:

$$\frac{x+a}{cz} = \frac{5+1}{0.0805 \times 48} = 1.552; \quad \frac{x-a}{cz} = \frac{5-1}{0.0805 \times 48} = 1.035$$

$$\frac{W}{W_o} = \sqrt{\frac{(0.97183 - 0.85673)}{2} \cdot \frac{(1+1)}{2}} = 0.239$$

Rectangular, Rounded-Entrance Apertures

Given: A rounded, rectangular source W_o = 3000 fpm; -1 in. $\leqslant x \leqslant +1$ in. (a = 1 in.); -3 in. $\leqslant y \leqslant +3$ in. (b = 3 in.).
Required: Same information as before.
Solution: (a) In Fig. 2 use $z/2a = 24$, $z/2b = 8$.
Then $W_c/W_o = 0.46$ and $W_c = 1380$ fpm.
(b) In Fig. 3 use $z/2a = 24$, $z/2b = 8$.
Then $E = 4.32$.
(c) This estimate is best accomplished with the aid of the following generalization of Equation 2:

$$\frac{W}{W_o} = \sqrt{\left\{\frac{\operatorname{erf}\frac{x+a}{cz} - \operatorname{erf}\frac{x-a}{cz}}{2}\right\}\left\{\frac{\operatorname{erf}\frac{y+b}{cz} - \operatorname{erf}\frac{y-b}{cz}}{2}\right\}} \quad \ldots\ldots (4)$$

Since:

$$\frac{x+a}{cz} = 1.552; \quad \frac{x-a}{cz} = 1.035$$

$$\frac{y+b}{cz} = 2.070; \quad \frac{y-b}{cz} = 0.517$$

$$\frac{W}{W_o} = \sqrt{\left\{\frac{0.97183 - 0.85673}{2}\right\}\left\{\frac{0.99658 - 0.53531}{2}\right\}} = 0.115$$

Therefore, $W = 346$ fpm.

Circular, Rounded-Entrance Orifices

Given: A rounded-entrance, circular source. $W_o = 3000$ fpm. Inside diameter: 0.9 in.

Required: (*a*) The peak velocity 9 in. from the source.
(*b*) The entrainment ratio 9 in. from the source.
(*c*) The velocity 9 in. from the plane of the source, and 1.08 in. off the jet axis.

Solution: As an adequate short-cut, circular jets are approximated by square jets of equal area, and radial velocity profiles are mimicked by the corresponding profiles along a principal axis of the square. (*a*) $(2a)^2 = (\pi/4)D^2$ Therefore, $2a = 0.795$ in. $z/2a = 11.3$; $z/2b = 11.3$.
From Fig. 2 $W/W_o = 0.565$; $W = 1700$ fpm.
(*b*) From Fig. 3 with $z/2a = 11.3$,
$z/2b = 11.3$, $E = 3.45$.
(*c*) For the off-axis velocity:

$$\frac{W}{W_o} = \sqrt{\left\{\frac{\operatorname{erf}\frac{r+a}{cz} - \operatorname{erf}\frac{r-a}{cz}}{2}\right\}\operatorname{erf}\frac{a}{cz}} \quad \ldots\ldots\ldots\ldots (5)$$

$$= \sqrt{\left\{\frac{0.99607 - 0.81720}{2}\right\}\{0.96497\}} = 0.294$$

$$W = 782 \text{ fpm}$$

Sharp-Edged Apertures: Figs. 1, 2 and 3 pertain to jet sources with rounded nozzle contours. With such sources the peak source velocity, W_o, persists over most of the source cross-section, and the contraction coefficient and discharge coefficient are close to unity (the value assumed for the charts). With other sources, having edges of varying degrees of sharpness, the contraction coefficient (and also the discharge coefficient) ranges from the minimum of 0.61 upwards toward 1.00.[6,8] To account for this type of deviation from ideal conditions, the empirical correction method used by others is adopted and each dimension of the original source is multiplied by the factor

$$\sqrt{C_d}.$$

The effective propagation area of the source is thus equated to its effective flow area. In Figs. 1, 2 and 3, therefore, a and b are replaced by a' and b' determined as follows:

$$a' = \sqrt{C_d}\, a; \quad b' = \sqrt{C_d}\, b.$$

COMPARISONS OF THEORY AND EXPERIMENT

Some of the analytical solutions of Equation 1 will be considered to determine how they agree with experiment. All comparisons will be made using the single value of $c = 0.0805$.

Infinite, Rounded-Entrance Slots: In this case the problem is two-dimensional, since the flow characteristics must be independent of y. Equation 1 reduces to:

$$\frac{\partial \rho W^2}{\partial (z^2)} = \frac{c^2}{4} \cdot \frac{\partial^2 \rho W^2}{\partial x^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

The methods of Reference 9, Chapter II, readily yield for this case:

$$\frac{W}{W_o}=\sqrt{\left\{\frac{\operatorname{erf}\frac{a+x}{cz}-\operatorname{erf}\frac{x-a}{cz}}{2}\right\}} \quad \ldots\ldots\ldots\ldots (7)$$

When $a/cz>>1$, this formula reduces to the following:

$$\frac{W}{W_o}=\sqrt{\frac{1-\operatorname{erf}\frac{x-a}{cz}}{2}} \quad \ldots\ldots\ldots\ldots (8)$$

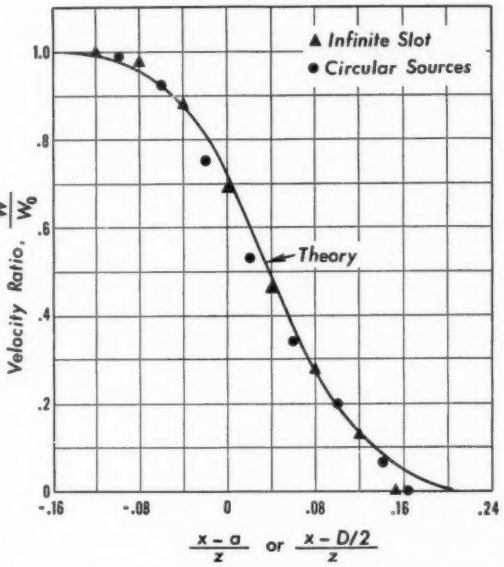


FIG. 4. COMPARISON OF THEORY AND EXPERIMENT FOR VELOCITY DISTRIBUTIONS NEAR ROUNDED CIRCULAR SOURCES

Thus W/W_o becomes a function of the single variable $(x-a)/z$ in the *near* field. Fig. 4 compares this result with the long-slot experiments of Albertson et al.[10]

When $a/cz<<1$, the formula becomes:

$$\frac{W}{W_o}=\sqrt{\frac{2a}{cz\sqrt{\pi}}}\exp\left(\frac{-x^2}{2c^2z^2}\right) \quad \ldots\ldots\ldots\ldots (9)$$

giving the Gaussian Error profile, with an attenuation proportional to the

inverse square root of distance from the source. Fig. 5 shows how Equation 9 agrees with measurements in the *far* field, as reported by Reference 10.

Along the jet center-plane, Equation 7 becomes:

$$\frac{W}{W_o} = \sqrt{\text{erf}\,\frac{a}{cz}} \qquad (10)$$

This is a simple and continuous expression for the peak velocity, which is valid from the plane of the source ($z = 0$) out to infinity. Fig. 6 compares this analytical result with the data of Reference 10.

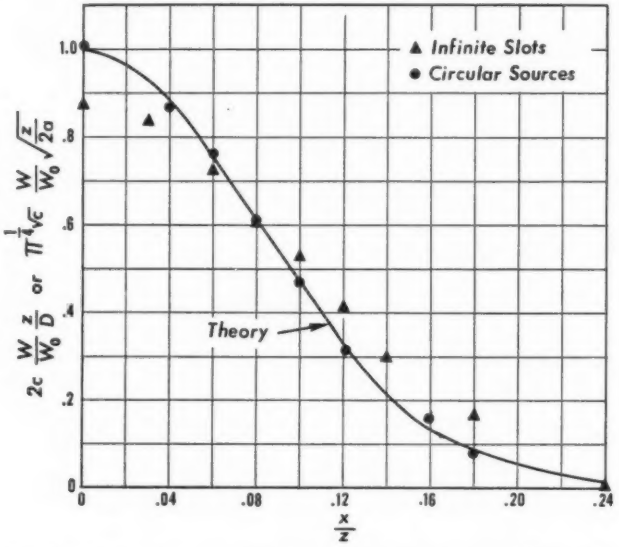


FIG. 5. COMPARISON OF THEORY AND EXPERIMENT FOR VELOCITY DISTRIBUTION AT A DISTANCE FROM ROUNDED INFINITE SLOTS AND ROUNDED CIRCULAR SOURCES

Entrainment ratios for the infinite slot can be found by integration of Equation 7. The results of such integration are incorporated in Fig. 3. In Table 1 they are compared with the findings of Albertson et al.

Discrepancies between theory and experiment only become important at some distance from the source, where better agreement would be achieved with a value of $c = 0.1$. Moreover, these discrepancies seem less serious when differences between observers are examined. Equation 11 due to Becher, for example, indicates $c = 0.087$.

Rectangular Apertures (Rounded and Sharp-Edged): The general theory of Reference 9, Chapter I, shows that the solution for a rectangular source of dimensions $a \times b$ can be regarded as the product of the solution for an infinite slot of width $2a$ and the solution for an infinite slot of width $2b$. Thus Equation 4 is obtained from Equation 2. This theoretical conclusion is strikingly con-

firmed by the following empirical equations of Becher, which are applicable in the *far* field.

Infinite Slots (rounded):

$$\frac{W_c}{W_o} = 2.55\sqrt{\frac{2a}{z}} \quad \ldots\ldots\ldots\ldots\ldots (11)$$

Rectangular Apertures (rounded):

$$\frac{W_c}{W_o} = 6.50\left\{1+0.13\left(\frac{a}{b}\right)^2\right\}\frac{\sqrt{(2a)(2b)}}{z} \quad \ldots\ldots\ldots\ldots (12)$$

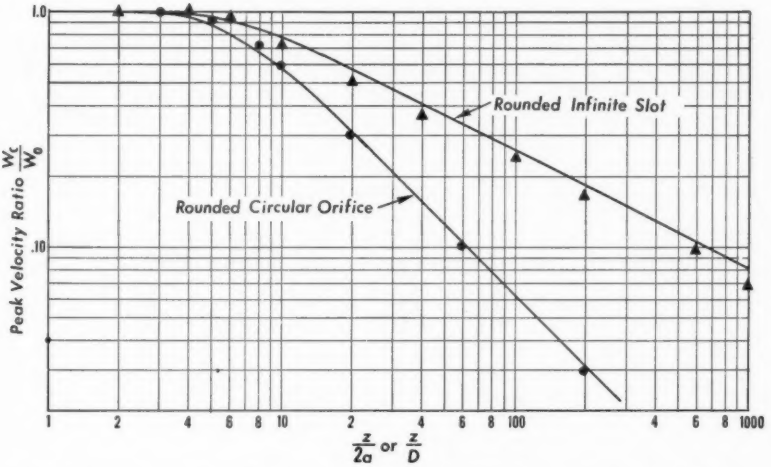


FIG. 6. COMPARISON OF THEORY (SOLID LINE) AND EXPERIMENT FOR CENTER, OR PEAK, VELOCITIES FROM ROUNDED INFINITE SLOTS AND ROUNDED CIRCULAR SOURCES

Except for the term $0.13(a/b)^2$, which is of no importance when $a/b < \frac{1}{2}$, and not of much importance in any event, the prediction is precisely confirmed.

An expression for the velocity along the axis of a rectangular source is obtained directly from Equation 4. Thus:

$$\frac{W_c}{W_o} = \sqrt{\operatorname{erf}\frac{a}{cz}\cdot\operatorname{erf}\frac{b}{cz}} \quad \ldots\ldots\ldots\ldots (13)$$

This formula is compared with the experiments of Becher in Fig. 7. The solutions exhibit first the constant *core* velocity, then the inverse square root behavior, and finally, the characteristic reciprocal attenuation of the *far* field. The *transition* points appear at nearly the right distances from the sources.

Entrainment ratios for sharp-edged, rectangular sources were obtained by Tuve, Priester and Wright.[11] Table 2 compares these data with values obtained

TABLE 1—ENTRAINMENT RATIOS FOR THE INFINITE SLOT

z/2a	ENTRAINMENT RATIOS	
	THEORY	EXPERIMENT[10]
1	1.08	1.08
2	1.15	1.16
3	1.24	1.24
4	1.32	1.32
5	1.40	1.40
10	1.78	1.98
20	2.46	2.77
40	3.40	3.92
80	4.78	5.54
110	5.60	6.50

by integration of Equation 4, and embodied in Fig. 3. A constant discharge coefficient of 0.725 was used in the calculations in accordance with the method outlined in the preceding section on Use of Computation Charts. The agreement between theory and experiment is probably within the experimental error.

Circular Orifices (rounded): In a discussion of circular jets, distinction must be made between true solutions of Reichardt's equation (Equation 1) and the equivalent-square approximation recommended by the author for design use. True solutions for finite circular sources were first compared with experiment by Alexander.[2] The success of his work, and its practical value, encouraged the present writer to undertake extensions to non-circular outlets. Although the

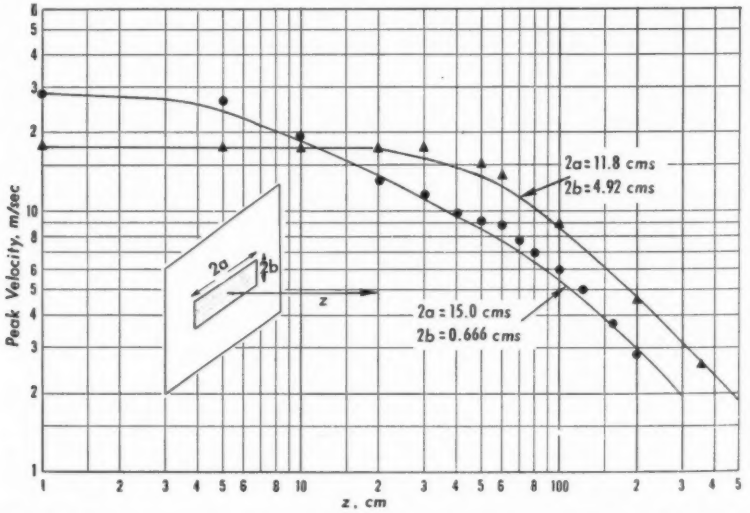


FIG. 7. COMPARISON OF THEORY (SOLID LINE) AND EXPERIMENT FOR CENTER, OR PEAK, VELOCITIES FROM ROUNDED, RECTANGULAR JET SOURCES

(*Note:* 1m/sec = 200 fpm; 10 cms = 4 in.)

contents of the next few paragraphs add nothing fundamental beyond what Alexander has already established, they do add certain new theoretical results and further experimental confirmation.

The analytical solution for a circular jet can be written directly from Reference 9, p. 220. Thus:

$$\left(\frac{W}{W_0}\right)^2 = \frac{2}{c^2 z^2} \exp\left(\frac{-r^2}{c^2 z^2}\right) \int_0^R \exp\left(\frac{-r'^2}{c^2 z^2}\right) J_0\left(\frac{i2rr'}{c^2 z^2}\right) r'dr' \quad \ldots\ldots (14)$$

In general, the indicated integral cannot be evaluated in closed form, but must be expressed in terms of infinite series, such as those given in the Appendix of this paper. However, simple expressions do exist in the following special cases.

(*a*) *Close to the Source:*

$$\left(\frac{W}{W_0}\right)^2 = \frac{1}{2}\left\{\operatorname{erf}\frac{r}{cz} - \operatorname{erf}\frac{r-\frac{D}{2}}{cz}\right\} - \frac{1}{\sqrt{8\pi}} \frac{cz}{r} \exp\left(\frac{-r}{cz}\right)\left\{\exp\left(\frac{D}{2cz}\right) - 1\right\} \quad \ldots (15)$$

TABLE 2—ENTRAINMENT RATIOS FOR SHARP-EDGED RECTANGULAR SOURCES

OUTLET DIMENSIONS IN.	ASPECT RATIO	ENTRAINMENT RATIO AT 9 FT FROM SOURCE[a]	
		THEORY	EXPERIMENT[11]
Fixed Area			
8.48 x 8.48	1.0	4.50	4.36
10.3 x 7	1.47	4.50	4.93
12 x 6	2.0	4.55	4.25
14.4 x 5	2.9	4.60	4.71
18 x 4	4.5	4.70	4.30
24 x 3	8.0	5.00	4.81
Fixed Length			
24 x 3	8.0	5.00	4.81
24 x 2	12.0	5.95	6.61
24 x 1	24.0	8.52	8.12
24 x ½	48.0	12.05	11.43

[a] The source velocity for these data was approximately 1100 fpm.

This equation has a maximum error less than 0.28 cz/r as long as the absolute value of $(1-D/2r)$ is not greater than one.

(*b*) *Far from the Source:*

$$\frac{W}{W_0} = \frac{D}{2cz} \exp\left(\frac{-r^2}{2c^2 z^2}\right) \quad \ldots\ldots\ldots\ldots (16)$$

This equation constituted the original basis for the Reichardt equation.

(*c*) *On the Jet Axis:*

$$\left(\frac{W}{W_0}\right)^2 = 1 - \exp\left(\frac{-D^2}{4c^2 z^2}\right) \quad \ldots\ldots\ldots\ldots (17)$$

This expression is exact. It was given by Alexander.

(*d*) *On the Cylinder containing the Source* ($r = D/2$):

$$\left(\frac{W}{W_0}\right)^2 = \frac{1}{2}\left\{1 - \exp\left(\frac{-D^2}{2c^2z^2}\right)\right\} J_0\left(\frac{iD^2}{2c^2z^2}\right) \quad \ldots\ldots\ldots\ldots (18)$$

This expression is exact. It is a new result.

Comparing first Equation 15 with Equation 8, it is noted that the velocity patterns of both the circular jet and the infinite slot should be similar near the source. This conclusion is substantiated by the experimental data of Albertson et al,[10] as shown in Fig. 4. However, the original, unaveraged data for the circular source show that a trend with z begins sooner in this case, than in the case of the infinite slot. On the cylinder $r = D/2$, Equation 18 reproduces this trend quantitatively.

Fig. 5 shows the expected good agreement of Equation 16 with experiment.

To validate the equivalent-square approximation, center velocities were computed with the aid of Fig. 2, and compared with experimental results in Fig. 6.

Finally, equivalent-square entrainment ratios are compared with the findings of Albertson et al in Table 3. The agreement is all that might be desired.

TABLE 3—ENTRAINMENT RATIOS FOR CIRCULAR JETS

	ENTRAINMENT RATIOS	
z/D	THEORY[a]	EXPERIMENT[10]
1	1.18	1.14
2	1.38	1.32
3	1.60	1.56
5	2.09	2.17
10	3.42	3.20
20	6.6	6.4
40	13.0	12.8
80	25.3	25.6
93	29.0	29.8

[a] As represented by the equivalent-square approximation.

CONCLUSIONS

1. With varying degrees of success, but with an accuracy sufficient for almost all engineering purposes, the free-turbulence equation of Reichardt correlates the performance of free jets. A value of 0.0805 for the constant c in Equation 1 provides satisfactory agreement with experiment.

2. A rectangular source of dimensions $a \times b$ behaves like the product of two infinite slot sources, one of width $2a$, the other of width $2b$. As a result, easily-used nomograms for peak velocity and entrainment ratio (Figs. 2 and 3) can be constructed. These nomograms are valid for sources of all aspect ratios.

3. The peak velocities and entrainment ratios of a circular jet are well approximated by those for a square jet of equal source area. Thus Figs. 2 and 3 may be employed for circular jets.

ACKNOWLEDGMENT

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APPENDIX

Series Expansions for the Velocity from a Circular Source

The integration in Equation 14 can be made by parts with the aid of the following well-known recurrence formula for the Bessel functions:

$$x^p J_{p-1}(\omega x) = \frac{1}{\omega} \cdot \frac{d}{dx}\left\{x^p J_p(\omega x)\right\} \quad \ldots\ldots\ldots\ldots \quad (19)$$

Since $J_p(0) = 0$ for all $p \geq 1$, the following series expansion is obtained:

$$\left(\frac{W}{W_0}\right)^2 = \exp\left\{\frac{-r^2}{c^2 z^2}\left(1+\frac{D^2}{4r^2}\right)\right\} \sum_{p=1}^{\infty} \left(\frac{D}{2r}\right)^p J_p\left(\frac{irD}{c^2 z^2}\right) \quad \ldots\ldots\ldots \quad (20)$$

Another expansion, suitable for small r, can be found by means of the generating function for the Bessel functions. Thus:

$$\exp\left\{\frac{x}{2}\left(t-\frac{1}{t}\right)\right\} = \sum_{p=-\infty}^{+\infty} J_p(x) t^p = \sum_{p=0}^{\infty} J_p(x) t^p + \sum_{p=1}^{\infty} J_{-p}(x) t^{-p} \quad \ldots\ldots \quad (21)$$

Since, for integral values of p:

$$J_{-p}(x) = (-1)^p \cdot J_p(x) \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (22)$$

then:

$$\sum_{p=1}^{\infty} J_p(x)\, t^p = \exp\left\{\frac{x}{2}\left(t - \frac{1}{t}\right)\right\} - \sum_{p=1}^{\infty} (-1)^p \cdot J_p(x)\, t^{-p} - J_o(x) \quad \ldots \quad (23)$$

When Equation 23 is used in Equation 20, a second series is found.

$$\left(\frac{W}{W_o}\right)^2 = 1 - \exp\left\{-\frac{r^2}{c^2 z^2}\left(1 + \frac{D^2}{4r^2}\right)\right\} \sum_{p=0}^{\infty} \left(\frac{2r}{D}\right)^p \cdot \frac{J_p\left(\frac{irD}{c^2 z^2}\right)}{(i)^p} \quad \ldots \ldots \quad (24)$$

Finally, adding together one-half of each of the above two series, a third expansion, useful near $r = D/2$, is obtained:

$$\left(\frac{W}{W_o}\right)^2 = \frac{1}{2} - \frac{1}{2} \exp\left\{-\frac{r^2}{c^2 z^2}\left(1 + \frac{D^2}{4r^2}\right)\right\} \left[J_o\left(\frac{irD}{c^2 z^2}\right) + \sum_{p=1}^{\infty} \left\{\left(\frac{D}{2r}\right)^p - \left(\frac{2r}{D}\right)^p\right\} \frac{J_p\left(\frac{irD}{c^2 z^2}\right)}{i^p}\right] \quad \ldots \quad (25)$$

When $r = D/2$, this expression reduces to Equation 18.

1515

No. 1515

FILTRATION OF MICROORGANISMS FROM AIR BY GLASS FIBER MEDIA

By H. M. Decker,* J. B. Harstad,** F. J. Piper,** and M. E. Wilson,*
Frederick, Md.

THERE IS AN increasing need for highly effective air filtration systems in hospitals, certain industrial installations, and in research laboratories in which it is required that air be relatively free of microorganisms before, during, or after a process. A survey of laboratory acquired infections in the United States, conducted by Sulkin and Pike,[1] shows a total of 1,342 infections presumably acquired in the laboratory. Concomitantly, it has been found that common laboratory techniques such as pipetting from vessels or mixing in blending devices permit the escape of organisms into room air, with subsequent danger to laboratory personnel. Only recently has the bacteriologist become aware of dangers that may be inherent in many laboratory operations previously considered safe.

This paper gives a brief review of work that has been reported regarding the use of glass fiber media in removing microorganisms from air and describes recent work involving the use of new types of filter media.

Review of Past Progress

A paper concerning the removal of bacteria and bacteriophage from air by electrostatic precipitators and spun glass filter pads was presented by the author in 1951.[2] It was reported that two ½-in. pads of spun glass having an average fiber diameter of 1.28 microns showed an average arrestance of 99 percent against clouds of *Serratia indica* and *Escherichia coli* T-3 bacteriophage atomized into the air stream by means of an all-glass direct-spray, peripheral air jet atomizer designated in Reference 3 as a Chicago type nebulizer because it was patterned after one of those used at the University of Chicago Toxicity Laboratories. Biological material from this nebulizer is not always unicellular, because agglomeration of organisms may occur during or after release of the aerosol. *S indica* is a harmless, elongated bacterial organism about one micron in length and one-half micron in thickness. *E coli* T-3 bacteriophage is a harmless virus organism, somewhat spherical in shape, which ranges in size from 0.02 to 0.05 microns. Since publication of the 1951 paper[2] reporting the use of spun glass filter media, such media have been found useful not only in general

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[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Swampscott, Mass., June 1954.

exhaust systems, but also in specialized apparatus and equipment such as bacteriological work cabinets,[4] shaking machines, and animal cages (Figs. 1, 2, and 3).

Because of the possible creation of a biological aerosol while changing or removing contaminated filters when the resistance becomes excessive, bacteriological safety cabinets were designed and fabricated with a spun glass filter system which may be decontaminated in place prior to removal of the filter. The addition of three 750-watt strip heaters to the filter system permits heating

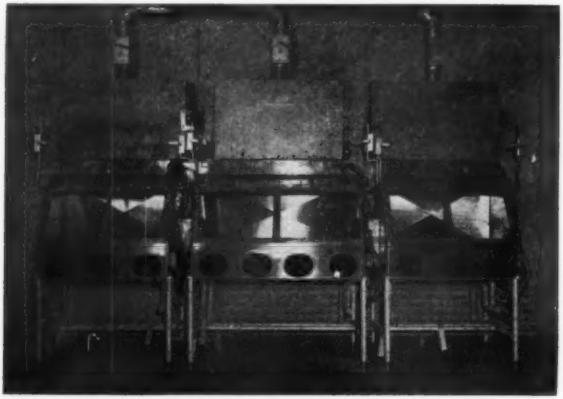


FIG. 1. BACTERIOLOGICAL SAFETY CABINETS

the filter interior to 200 C (392 F). Time temperature studies have shown that after the interior of the filter has reached 200 C, an exposure of 20 min is sufficient to destroy very resistant spore-forming bacterial organisms such as *Bacillus globigii*. Infection of personnel while changing filters is therefore eliminated.

It has been reported by Leavell and Amoss[5] that brucella, the causative organism of brucellosis, can be isolated from the feces and urine of infected

GLOSSARY

bacteria—a widely distributed group of one-celled microorganisms, some of which are disease producing.

bacteriophage—a virus organism which is a parasite of bacteria.

blender—mechanism used to mix organisms into solution.

liquid impinger—a sampling device used to collect organisms from the air by passing air through a liquid confined in a vessel.

pipetting—the act of drawing and releasing a liquid through a narrow glass tube.

shaking machine—a mechanical device used for shaking flasks containing bacteria in a liquid growth medium.

sieve sampler—a sampling device used to collect organisms in the air by impingement of the organisms onto a growth medium.

spore forming bacteria—bacterial organisms which develop a tough investing cell wall, and possess great tenacity of life.

viruses—extremely microscopic agents, some of which cause disease.

animals. This naturally indicates the possibility of airborne infection from infectious animal cages. To minimize this danger, several research institutions are ventilating the air through spun glass as it enters and leaves the cage. Use of the glass filter serves several purposes: (1) provides the animals with clean air; (2) lowers the possibility of cross-contamination between cages; and (3) in the event of a failure of the mechanical air supply system, contaminated air does not escape, but the animals still receive enough circulation of the air through the filters to prevent suffocation. Fig. 3 shows a photograph of the ventilated animal cage.

Another adaptation of spun glass has been its application to shaking machines. To biologists and bacteriologists, the breakage of culture flasks on a mechanical shaker is a problem which becomes a hazard in laboratories working with pathogenic agents. Loss of plugs from the culture flasks being shaken has also been reported. The possibility of accidentally dropping a flask during transfer must not be overlooked. To avoid the dangers from such accidents, a shaking machine container has been constructed to completely enclose the flask, and yet provide sufficient air through a spun glass filter for satisfactory bacterial growth. Fig. 4 illustrates this type of container; various modifications are possible.

The data and application of glass filter media presented before have been confined to spun glass pads varying in thickness from ½ to 1 in. As reported

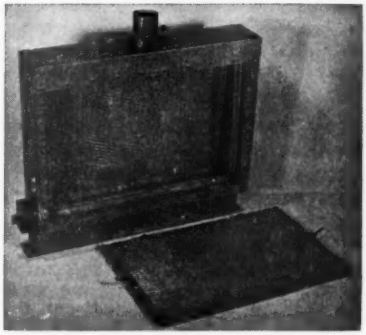


FIG. 2. BACTERIOLOGICAL SAFETY CABINET SPUN GLASS FILTER UNIT

in earlier work, two ½-in. pads of 1.28 micron size spun glass removed at a linear air flow of 20 fpm, an average of 99 percent of the bacteria and 99 percent of the virus organisms. It was felt that efficiencies of this range were entirely adequate for exhausting air from rooms where one would expect to find only a few infectious microorganisms. Actually, none should be released to the room air. The installation and use of the filters should be regarded as a safeguard against infection of personnel in the surrounding community in the event of an accident and ultimate release to the general building exhaust system.

Exhaust air filters with arrestance efficiencies in excess of 99 percent should be sought for specialized apparatus in which significantly infectious bacterial aerosols are accidentally or deliberately created, and for rooms in which infec-

tious organisms are handled in unusual amounts. At present, air from such locations is often passed through an air incinerator.

CURRENT DEVELOPMENTS

A new glass fiber filter paper originally developed jointly by the Department of the Navy and the National Bureau of Standards is now being produced commercially. This filter media is as thin as coarse paper (10 mils). The fibers are known commercially as type E Glass Micro Fibers. They have a melting point of 1450 F and an average diameter of 0.5 to 0.75 microns. A second new type of filter paper is being developed by a research and engineering organization. This second type is composed of a mixture of glass fibers and asbestos fibers. Figs. 5, 6 and 7 show the fibers as they appear in the 1.28 micron size spun glass filter pads and in the two new glass filter papers. All three have

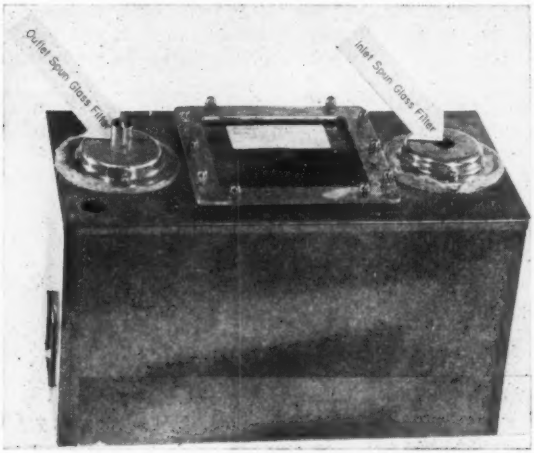


FIG. 3. VENTILATED ANIMAL CAGE



FIG. 4. SHAKING MACHINE CONTAINER

been tested to determine comparative efficiencies in the removal of bacteria from an air stream. The 1951 A.S.H.V.E. paper[2] reported an average arrestance of 99 percent of *S indica* and *E coli* bacteriophage T-3 when using two ½-in. spun glass filter pads at a linear air flow of 20 fpm. Since that time, glass paper filters have been evaluated.

EXPERIMENTAL METHODS

S indica was used to evaluate the efficiency of the filter papers. The test equipment is illustrated in Fig. 8. The organisms were nebulized into a cloud chamber where the cloud of bacteria was mixed with air, then passed into a pre-filter sampling chamber, through the filter at face velocities of 10 and 20 fpm into a post-filter sampling chamber, and finally exhausted by means of a blower to the outside air.

The air was sampled in front of the filter by means of liquid impingers constructed with a critical orifice which permitted air to be drawn through the

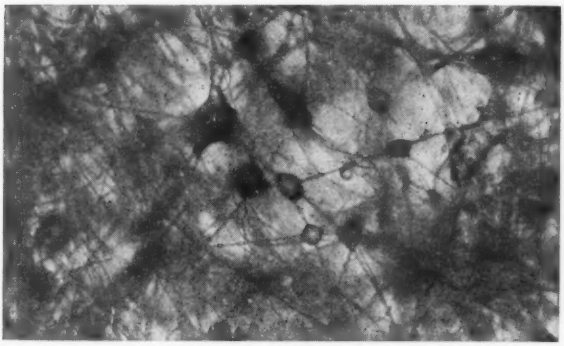


FIG. 5. PF 105 SPUN GLASS FILTER MEDIA

Fiber diameter (average) 1.28 microns

collecting medium at approximately 0.5 cfm. The collecting medium in the impingers consisted of 20 ml nutrient broth and 6 to 8 drops of olive oil. One-tenth ml of the sample was streaked on a corn steep agar plate. In addition, one ml samples from the liquid impingers were serially diluted and one-tenth ml samples of the dilutions were streaked on plates for incubation and counting. Sieve samplers containing corn steep agar petri plates were used to sample air after it had passed through the filter.

RESULTS

The efficiency of the filter paper was determined by sampling the cloud concentration before and after the filter. The results of the test are shown in Table 1. At an air flow of 20 linear fpm, penetration of *S indica* through the type E glass filter paper was two organisms from each 100 million test organisms recovered in front of the filter. At an air flow of 10 linear fpm, the penetration was 1 organism. With the asbestos glass paper, at an air flow of 20 linear fpm,

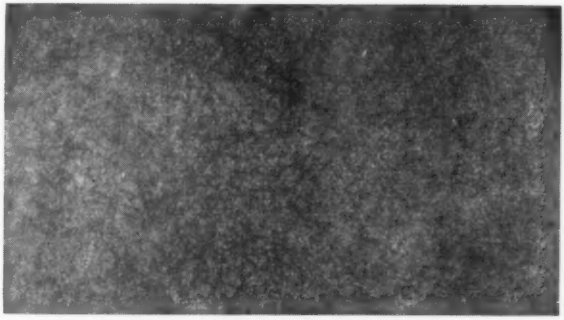


FIG. 6. TYPE E GLASS FILTER PAPER

Fiber diameter range 0.2 to 1.5 microns. Average diameter 0.5 to 0.75 microns. Melting point 1450 F



FIG. 7. ASBESTOS GLASS FILTER PAPER

the penetration was 28 organisms per 100 million test organisms. When the air flow through the asbestos glass paper was reduced to 10 linear fpm, the penetration increased to 140 organisms. The increase in penetration from 28 to 140 probably results from the lesser impingement of the organisms on the fibers at the lower velocities.

CONCLUSIONS

The filtration arrestance of the three filter media is very high and indicates possibility for wide practical application of these filters. Results of the authors' earlier work showed that two ½-in. pads of 1.28 micron size spun glass at a linear air flow of 20 fpm removed an average of 99 percent of the bacteria and virus organisms. It was felt that this type of air filtration system would be satisfactory for general building exhaust as in hospitals and in industrial concerns. However, in specialized circumstances such as in the case of some research institutions where organisms may be handled in large numbers or in apparatus in which significantly infectious bacterial aerosols are accidentally or deliberately created, a greater arrestance is necessary.

A highly efficient mineral filter paper is now available that can be easily used at high temperature. It is fire resistant, does not disintegrate when wet, and can be biologically decontaminated by heat.

TABLE 1—EFFICIENCY OF GLASS AND ASBESTOS-GLASS FILTERS IN REMOVAL OF *S Indica* FROM AN AIR STREAM

FILTER MATERIAL EVALUATED	AIR FLOW LINEAR FPM	RESISTANCE IN. WATER	NUMBER OF FILTERS[a] EVALUATED	PENETRATION PER 100 MILLION ORGANISMS RECOVERED BEFORE FILTER
Glass type E	20	3.2	7	2
filter paper	10	1.5	5	1
Asbestos Glass	20	3.1	5	28
paper	10	1.5	5	140

[a] Each filter was tested a minimum of 10 times.

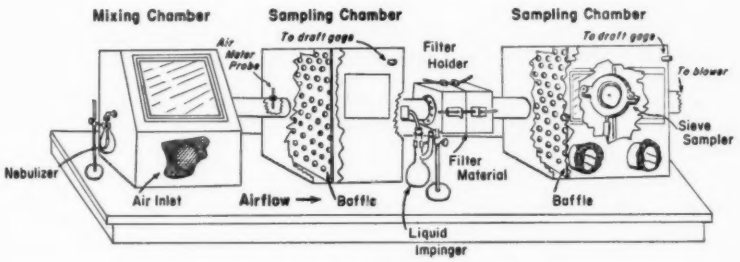


FIG. 8. AIR FILTRATION TEST UNIT

A non-combustible filter frame for the all-glass fiber filter is being developed commercially (Fig. 9). The filter made of asbestos fibers and glass fibers is also approaching commercial availability. Industrial installations, hospitals and research laboratories which require removal of biological, radiological or other particulates from an air stream, now have access to a highly effective, fire and chemical resistant (except for hydrogen fluoride and alkalies) filter. Frequently, such a filtration system may be used in lieu of a costly incinerator.

REFERENCES

1. Survey of Laboratory Acquired Infections, by E. S. Sulkin and R. M. Pike (*American Journal of Public Health*, 1951, Vol. 41, pp. 769-781).

2. Removal of Bacteria and Bacteriophage from the Air by Electrostatic Precipitators and Spun Glass Filter Pads, by H. M. Decker, F. A. Geile, H. E. Moorman, C. A. Glick (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, October 1951, pp. 125-129).

3. *Experimental Airborne Infection*, by Theodor Rosebury (Williams and Wilkins, 1947, p. 18).

4. Spun Glass Air Filters for Bacteriological Cabinets, Animal Cages, and Shaking Machine Containers, by H. M. Decker, F. A. Geile, J. B. Harstad, N. H. Gross

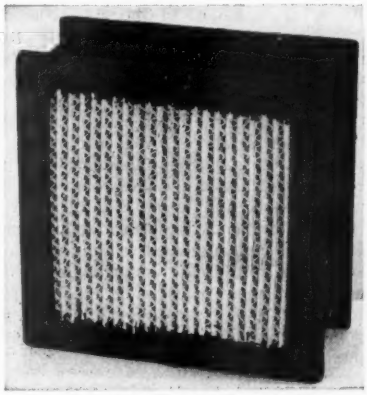


FIG. 9. ALL GLASS FILTER UNIT

(*Journal of Bacteriology*, Vol. 63, No. 3, 1952, pp. 377-384).

5. Brucella Infection Case Report. Cultivation of Brucella from Bile, by H. R. Leavell and H. L. Amoss (*American Journal of Medical Science* 1931, Vol. 181, pp. 96-102).

DISCUSSION

S. F. DUNCAN, Los Angeles, Calif. (WRITTEN): This paper is a distinct contribution to the literature on air cleaning. The authors show considerable restraint in reporting their tests since it takes painstaking care to evaluate the performance of filter media operating at the efficiencies found.

It is a bit surprising to see that the Type E glass filter paper tended to improve in collection of *S indica* as the velocity decreased, while the asbestos glass paper reacted oppositely. The authors attribute the increase in penetration at low velocity on the asbestos glass paper to the lower tendency for impingement at the lower velocity. Another explanation for the observed change in penetration on the asbestos glass paper could be the variable density of the material as shown in Fig. 7. Areas of low pack density would, at low average velocity, carry the bulk of the air, and hence give less opportunity for *S indica* to deposit on the fibers. At 20 fpm average face velocity more air will flow through the close packed areas and improve performance as noted.

The fact that *S indica* is close to the size of a 1 micron sphere puts it in the range where Brownian movement can influence the path of the particle. Also, 1 micron spheres have a very short stopping distance, and are difficult to impinge on any kind of a fiber. The combination of particle size and fiber size, together with the relatively constant penetration in the Type E glass fiber paper, leads toward the conclusion that particle collection is more influenced by diffusion than by velocity impingement. Hence the suggestion that the increase in pentration through the asbestos glass paper at low velocity is due to nonuniform paper density, is presented as a more likely explanation than that usually inferred by the wording of the authors' comment.

A. B. HUBBARD,* Bloomfield, N. J. (WRITTEN): The authors have provided valuable data for the guidance of laboratories dealing with hazardous microorganisms. They are to be commended for undertaking such a formidable task and for presenting the results so ably.

The numerous variables of filtration make it difficult to compare filters in ways easily visualized. An attempt is made here to compare the spun glass filter pads† with the Type E glass filter paper. One in. thick pads of 1.28 micron fiber size have a resistance of 20 fpm of 0.96 in. of water and are penetrated by one particle per hundred. Ten mil thick glass paper at the same velocity has a resistance of 3.2 in. of water but is penetrated by two particles per 100 million.

Now if one assumes that resistance increases linearly with thickness while penetration decreases linearly as thickness increases exponentially, it is possible to estimate what a thicker pad will do at the same resistance and also how thick a pad would have to be to show the same efficiency as the glass paper. It turns out that a 3.34 in. pad would pass 21 particles per 100 million at 3.2 in. of water and that a 3.85 in. pad should duplicate the 2 per 100 million performance at 3.7 inches of water resistance. Thus it can be seen that the glass paper does a filtering job that could be matched by spun glass only by resorting to awkward arrangements of very thick pads.

The 1951 paper shows nearly equal penetration by *S indica* one micron in length by one-half micron thick and *E coli* T-3 ranging in size from 0.02 to 0.05 microns. The same situation is implied in the present paper. In dust filtration one would expect the smaller organisms to penetrate a given filter at least 20 times as often

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† See Reference 2 of the paper.

as the larger ones do. It is likely that the nebulizer presents droplets of liquid to the filter having a larger size than either of the organisms carried by them. If this is the case, a different nebulizer might cause a difference collection. In that case one would look for a variation of efficiency with humidity of the air entering the test apparatus. Can hazardous microorganisms always be expected to appear in practice in liquid droplets of size comparable to those used in the tests? That is, can the test results be used literally to estimate practical performance?

R. L. KUEHNER, York, Pa. (WRITTEN): The very fine progress the authors have recorded in this and preceding papers on air filtration is noted. The incorporation of heating devices for post-disinfection of contaminated filters is a unique and a very necessary development for the designed applications. Recent patent art shows that effort is being made by others to accomplish the same end for more general air conditioning use. These efforts have been largely in the direction of chemical disinfection either by incorporating a disinfectant in the new filter, or by chemically treating the filter immediately before removal and discard. Would the authors care to comment on whether or not they feel that such end-decontamination is necessary or desirable for the general field of air conditioning? Have they any indicators that the heat, chemical, or other method shows the greatest promise?

At first glance, the pressure drop through the two filters seems prohibitively great for general use. Fig. 9 in the paper suggests that the filters are physically amenable to the accordian type mounting, similar to the cellulose-asbestos paper used in *A.E.C.* work. If so, practical pressure drop becomes a function of the amount of filter paper surface exposed, hence, overall size of the filter. Are data available on pressure drop per overall size of filter per volume of air, for example, 500 cfm?

I would also like to ask the authors' comments on several other problems in the general field of filter testing. In our work, we have found it desirable to use identical air sampling devices before and after the removal mechanism under test. In many cases this is not possible. For example, in the paper under discussion the number of bacteria before and after the filter probably dictated the use of the impinger and the sieve device. Space limitations and pressure differences frequently cause similar inconveniences. In such cases the comparative collective efficiencies of two dissimilar devices must be determined. Do the authors have or know of a simple, standard or generally accepted method for comparing efficiencies of air sample devices?

Secondly, on what basis is the decision made to use wet or dry suspensions, bacteria carried on dust or in droplet nuclei, or other inocula in filter rating. Also of what importance is the size of the inocula? Is there general agreement on these questions?

Finally, what are the general criteria applied in the selection of the specific organisms to be used in filter rating studies?

AUTHORS' CLOSURE (H. M. Decker): Replying to Mr. Duncan, the data presented for the Type E glass paper shows penetration of 2 organisms per 100 million organisms recovered before the filter at a linear airflow of 20 fpm and 1 organism at a linear airflow of 10 fpm. It is felt that this difference is not significant. The authors are of the opinion that one micron particles are more influenced by velocity change than Brownian movement. A review of literature and experimental data would indicate that the greater velocity was the prime cause for the variation in penetration.

With reference to the comments of Mr. Hubbard, the nebulizers used in the comparison of the spun glass filter pads** with the Type E glass filter paper were of the same type. Under the conditions tested, the nebulizer aerosolized particles in the range of one micron, however, it is realized that particle size will vary with different nebulizers, and with varying relative humidity. With reference to the question as to whether hazardous organisms can be expected to appear in liquid droplets of sizes

** See Reference 2 of the paper.

comparable to those used in the tests, the authors feel that one cannot expect exact standardization of a particle, however the test results can be used literally to estimate practical performance.

Discussing the remarks of R. L. Kuehner, chemical disinfection either by incorporating a disinfectant in the new filter or by chemically treating the filter immediately before removal and discard is not necessary for the general field of air conditioning. However, hospitals, research institutions, and any organization handling pathogenic organisms should decontaminate filters prior to changing. Chemical disinfectants require too great a time concentration factor for destruction of bacterial spores. It is felt that heat would show the greatest promise.

The authors realize that individual air sampling devices before and after the filter are the most desirable method. At the time the initial studies on the filters described in the paper were conducted, *Serratia indica*, a vegetative organism was used. Collection of low concentration of the test organism was limited to the sieve sampler. However, later tests with bacterial spores permitted the use of identical air sampling devices before and after the filters. The data collected show little variation with the original data presented in this paper.

The authors have and know of a simple standard and accepted method for comparing efficiencies of air sampling devices. However, discussion of this aspect does not come within the scope of this paper.

The decision to use wet or dry suspension, bacteria carried on dust or in droplet nuclei or other inocula is based on the ability to produce a uniform homogeneous aerosol cloud which is reproducible.

1516

No. 1516

EFFECTS OF ROOM SIZE AND NON-UNIFORMITY OF PANEL TEMPERATURE ON PANEL PERFORMANCE

By L. F. SCHUTRUM* AND J. D. VOURIS,† CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

SEVERAL YEARS ago, a special room was constructed at the A.S.H.V.E. Research Laboratory for the study of heat transfer within a panel heated space. This room, commonly referred to as the Environment Laboratory, is approximately 12 ft wide and 24½ ft long, and the ceiling is adjustable to any height up to 12 ft. Either the floor or the ceiling, or both, may be heated, and all six surfaces may be cooled to any temperature or combinations of temperatures desired. Cooled air can be introduced at six locations in the room to simulate infiltration. A detailed description of the construction and instrumentation of this room is given in an earlier paper.[1]

In December 1952, a paper[2] was published giving the results of tests made in the Environment Laboratory with a heated ceiling, uniform environment (all surfaces other than the heated ceiling at a uniform temperature), and with and without simulated infiltration. In June 1953, a second paper[3] was presented, giving the results of a similar series of tests made with a heated floor. In a more recent paper,[4] the modifying effects of non-uniform environment, room furnishings, and carpeting were discussed.

All of the results presented in previous papers were obtained in a room 24½ ft long and 12 ft wide with an 8 ft ceiling height, and with either the entire floor or ceiling heated. The room with these dimensions will hereafter be referred to as the *standard room*. The tests reported in this paper were made to determine the effects of room size, and heating only portions of the floor or ceiling area on panel heat output and room air temperature. All tests were made with the room unfurnished and unlighted.

EFFECT OF ROOM SIZE ON PANEL PERFORMANCE

Test Methods: Since the temperature of all six room surfaces could be controlled directly, it was experimentally expedient to fix these temperatures and observe the resulting dependent variables, room air temperature and panel output. As will be shown later, from these observed interrelationships of room air

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† Assistant Research Engineer, A.S.H.V.E. Research Laboratory.
[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1954.

temperature, panel output, and surface temperatures, the performance of a panel in a room of known thermal properties can be deduced for any combination of inside and outside air temperatures.

To determine the effect of room size on panel heat output and room air temperature, rooms of three sizes other than the *standard room* were tested. First, the ceiling of the *standard room* was raised to create a room $24\frac{1}{2} \times 12 \times 12$ ft high. Second, a smaller room 12 ft square with an 8 ft ceiling height was made by installing a temporary partition across the larger room. Although the temperature of this partition could not be directly controlled, heat flow meters and thermocouples located on the wall surfaces provided adequate information regarding its thermal conditions.

For each of these two room sizes, complete series of tests were made with various combinations of ceiling or floor panel surface temperatures, unheated room surface temperatures, and infiltration air quantity and temperature.

A few tests were also made in a $24\frac{1}{2} \times 12$ ft room with a 4 ft ceiling. This, of course, was not a practical room height, but provided the same geometrical proportions as a room $49 \times 24 \times 8$ ft high. However, a complete simulation of heat transfer was not obtained.

Effect of Room Size on Ceiling Panel Performance: The results of tests with ceiling panels in the rooms just described, are presented in Table 1. Column 1 of the table gives the general test conditions. Columns 3 to 8 inclusive show the observed temperatures of the six room surfaces. The area weighted average unheated surface temperature (AUST) for each test is given in column 9. Air changes per hour and the entering infiltration air temperatures are listed in columns 10 and 11.

The observed outputs per square foot of the heating panel are listed in column 12; and in column 13, listed for comparison, are the heat outputs which would have been obtained in the *standard room*. These latter panel outputs were predicted from heat flow and surface temperature relationship curves given in Figs. 1 and 4 of Reference 2, and are for a uniform environment having the same AUST, panel temperature, air changes per hour and infiltration air temperatures.

An analysis of the values in columns 12 and 13 indicates that the *average* ceiling panel output in the $12 \times 12 \times 8$ ft room was approximately 4 percent lower than in the *standard room* with comparable but uniform environmental conditions. The *maximum* difference was just over 7 percent. Similarly, the *average* panel output in the $24\frac{1}{2} \times 12 \times 12$ ft room was about 2 percent lower than in the *standard room* and the *maximum* difference was approximately 6 percent. The greater difference occurred with greater deviations from uniform environment.

The observed air temperatures taken at the center of the rooms 30 in. and 60 in. above the floor are given in columns 14 and 15 of Table 1. In the next two columns are comparable air temperatures determined from Figs. 7 and 9 of Reference 2. These are the air temperatures which would have prevailed in the *standard room* with a uniform environment having the same AUST, panel temperature, and infiltration air changes and temperature. Since all of the values in columns 16 and 17 are for uniform environment conditions, only the values for uniform environment tests in columns 14 and 15 are strictly comparable with them. However, good agreement exists for all tests and all are included in the following discussion of the air temperature data.

TABLE 1—OBSERVED CEILING PANEL OUTPUTS AND ROOM AIR TEMPERATURES FOR THREE ROOM SIZES
Calculated Values for a Room 24½ × 12 × 8 Ft High Shown for Comparison
Entire Ceiling Area Heated

General Test Conditions	Test No.	Room Surface Temperatures, F						AUST[a]	Infiltration		Panel Output Btu/(hr)(sq ft)		Room Air Temperature, F			
													Observed		Calculated[c]	
		Ceiling	Floor	North Wall	South Wall	East Wall	West Wall		Changes per hour	Temp F	Observed	Calculated[b]	30 in.	60 in.	30 in.	60 in.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Room Size 12 ft x 12 ft x 8 ft high																
Uniform Environment	332	119.8	65.3	65.8	65.4	65.5	65.4	65.4	—	—	59.5	59.5	67.3	68.9	67.5	69.8
	334	110.7	62.1	62.4	62.9	62.9	62.7	62.6	—	—	50.0	51.5	64.5	66.1	64.5	66.4
	339	120.3	64.4	64.8	64.8	64.8	64.7	64.7	3.0	19.9	61.5	64.0	59.0	62.1	55.8	60.2
One Wall Neutral (North)	287	120.2	64.7	77.3	65.6	65.5	65.6	67.4	—	—	58.5	59.0	68.6	72.0	69.5	71.6
	293	119.5	65.2	75.8	65.3	65.3	65.3	67.2	2.1	42.0	57.5	59.5	65.3	69.2	65.0	67.2
	294	120.4	65.3	75.6	65.4	65.5	65.5	67.3	3.0	40.3	59.0	61.5	63.6	67.8	62.8	65.2
	299	110.3	64.3	74.7	64.5	64.7	64.5	66.3	—	—	45.5	47.5	67.5	70.8	68.0	69.8
One Wall Neutral (North) One Wall Cold, (East)	288	120.2	65.3	71.6	65.4	35.4	65.5	61.1	—	—	63.0	65.0	61.4	66.8	63.4	65.8
	298	120.3	64.3	69.8	64.6	33.5	64.6	59.8	3.0	40.4	64.5	68.0	58.6	62.5	57.4	59.6
3 Simulated Windows[d] 2 Simulated Outside Walls, Floor, North, and West Walls Neutral	289	120.0	82.4	85.2	66.4	66.6	83.5	74.7	—	—	48.0	51.5	76.3	80.0	76.5	78.3
	292	119.9	78.7	82.2	68.1	68.2	79.9	72.9	2.1	42.0	51.5	55.0	70.2	74.5	69.5	71.8
Room Size 24½ ft x 12 ft x 12 ft high																
Uniform Temperature Environment	301	119.9	64.2	66.7	64.8	64.6	64.3	64.7	—	—	60.0	60.5	66.2	67.3	66.8	69.1
	302	119.8	54.7	57.6	55.3	54.7	55.1	55.2	—	—	69.0	70.0	56.7	57.8	57.7	60.4
	303	120.0	74.1	76.5	75.0	75.4	74.9	75.0	—	—	49.5	51.5	76.5	77.5	76.8	78.6
	304	119.6	64.4	67.1	65.3	65.3	65.2	65.3	1.5	40.4	60.0	61.5	62.1	64.2	64.5	66.8
	309	100.6	65.0	66.5	65.0	65.3	64.8	65.2	—	—	37.0	37.0	66.0	66.8	66.6	68.0
	310	120.4	64.3	66.5	64.9	65.0	64.8	64.9	1.9	40.6	62.5	63.5	60.5	62.4	63.4	65.7
One Long Cold Wall	314	119.7	64.1	64.8	64.9	34.5	64.4	56.9	1.5	20.4	69.0	70.5	51.5	54.5	54.9	58.0
	315	120.5	64.4	65.6	65.2	35.2	64.8	57.4	—	—	68.0	68.5	57.5	59.0	59.9	62.4
3 Simulated Windows[d] 2 Simulated Outside Walls, Floor, North, and West Walls Neutral	316	120.3	79.8	82.0	67.4	67.4	80.7	74.0	—	—	50.5	53.0	73.8	74.9	75.8	77.7
	317	120.9	71.8	75.1	67.3	67.3	74.7	69.6	1.5	20.7	57.0	60.5	61.0	64.8	65.6	69.1
Room Size 24½ ft x 12 ft x 4 ft high																
Uniform Environment	324	120.7	65.0	67.7	64 8	64.8	64.8	65.1	—	—	60.0	61.5	73.7	—	67.3	—
	325	110.9	64.9	66.9	64.5	64.4	64.5	64.9	—	—	48.0	49.0	71.8	—	66.7	—

[a] Area weighted average temperature for the floor and four walls.
[b] Determined from Figs. 1 and 4 of Reference 2 for the same AUST, panel temperature, infiltration temp. and number of air changes per hour in *standard room*.
[c] Determined from Figs. 7 and 9 of Reference 2 for the same AUST, panel temperature, infiltration temp. and number of changes per hour in *standard room*.
[d] Simulated windows 4 ft x 5 ft with temperatures approximately 33 F.

An analysis of the air temperatures at the 30-in. level in the 12 × 12 × 8 ft room and the *standard room* indicates an average difference (neglecting the sign) of less than a degree and a maximum difference of 3.2 deg. At the 60-in. level, the average difference was 1.6 deg; the maximum difference was 2.9 deg; and in all but two tests, the observed temperatures were higher in the smaller room.

A similar analysis of the air temperatures at the 30-in. level in the 24½ × 12 × 12 ft room and the *standard room* indicates an average difference

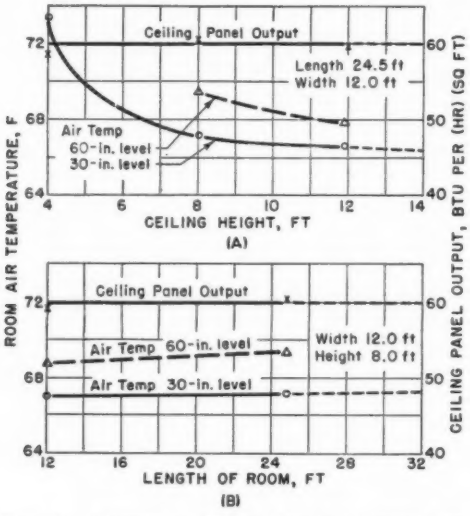


FIG. 1. EFFECT OF ROOM SIZE ON ROOM AIR TEMPERATURE AND CEILING PANEL OUTPUT

Uniform environment, no infiltration, ceiling 120 F AUST 65 F

of 2.0 deg and a maximum difference of 4.6 deg. At the 60-in. level, the average difference was 2.7 deg and the maximum difference was 4.3 deg. In all tests, the higher temperatures occurred in the *standard room.*

Fig. 1 indicates the trend of the effects of ceiling height and room length on unit output and room air temperature for ceiling panel heated rooms with a uniform environment and without infiltration. These curves were plotted from tests of a limited number of conditions and should be considered therefore as illustrative only.

For the rooms with lower ceilings, the room air temperature is higher. This is expected, since the ratio of heated surface area to unheated surface area is higher with lower ceilings. The air temperature in the 24½ ft long room is slightly higher than in the 12 ft long room for a constant ceiling height (Diagram B of Fig. 1). This also may be attributed in part to the higher ratio of heated to unheated areas. Higher ceiling panel outputs were expected in the rooms with lower air temperatures; however, since this was not observed, the increased

convection output of the ceiling panel must have been so small as to be masked by experimental errors.

The analysis given, taking into account the unavoidable inaccuracies in determining the AUST, seems to indicate that with ceiling panel systems, the effect of room size and ceiling height on panel output and room air temperature may be safely neglected in rooms of normal size and proportions.

Effect of Room Size on Floor Panel Performance: Table 2, which is similar in construction to Table 1, presents data obtained on floor panel heated rooms of sizes described. The values in columns 13, 16 and 17 were obtained from Figs.

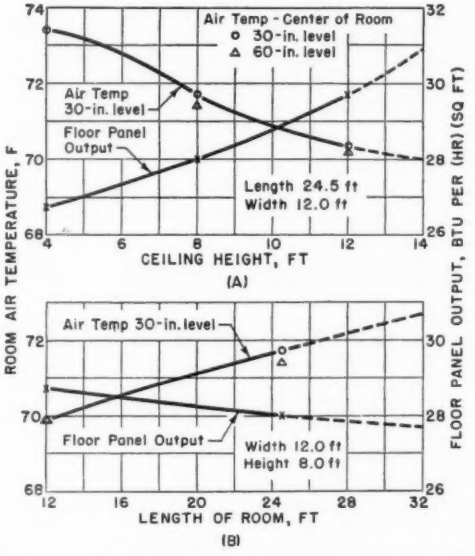


FIG. 2. EFFECT OF ROOM SIZE ON ROOM AIR TEMPERATURE AND FLOOR PANEL OUTPUT

Uniform environment, no infiltration, floor 85 F AUST 60 F

1, 4, 7, and 10 of Reference 3. The experimental air temperatures for the non-standard test rooms in columns 14 and 15, compared with those for the *standard room* in columns 16 and 17, show close agreement.

A comparison of the values in columns 12 and 13 shows the differences in the observed panel heat outputs and the heat outputs which would have been obtained in the *standard room* under the conditions specified in the footnotes of the table. The average difference (regardless of sign) between the observed heat outputs for the $12 \times 12 \times 8$ ft room and those for the *standard room* is 4 percent and the maximum difference is 8.2 percent.

The observed panel outputs for the $24\frac{1}{2} \times 12 \times 12$ ft room compared with those for the *standard room* show greater divergence. The average difference for all tests is approximately 8 percent; the average difference for the tests with

TABLE 2—OBSERVED FLOOR PANEL HEAT OUTPUTS AND ROOM AIR TEMPERATURES FOR THREE ROOM SIZES

Calculated Values for a Room 24½ × 12 × 8 Ft High Shown for Comparison

Entire Floor Area Heated

General Test Conditions	Test No.	Room Surface Temperatures, F						AUST[a]	Infiltration		Panel Output Btu/(hr)(sq ft)		Room Air Temperature, F			
													Observed		Calculated[c]	
		Floor	Ceiling	North Wall	South Wall	East Wall	West Wall		Changes per hour	Temp F	Observed	Calculated[b]	30 in.	60 in.	30 in.	60 in.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Room Size 12 ft x 12 ft x 8 ft high																
Uniform Environment	333	85.6	64.6	64.7	65.0	65.0	64.9	64.8	—	—	29.5	29.0	69.7	69.5	71.8	71.5
	335	90.6	65.0	64.4	65.1	65.1	65.2	65.0	—	—	38.0	36.5	71.4	71.8	73.7	73.2
	340	85.9	65.1	64.7	64.4	64.6	64.5	64.7	3.1	21.0	41.0	41.5	64.8	65.5	64.2	64.3
One Wall Neutral (North)	285	85.0	64.8	70.7	65.1	64.2	65.2	65.9	—	—	27.0	26.5	70.5	70.5	72.4	72.0
	295	84.9	65.0	69.5	64.6	64.4	64.6	65.5	3.0	40.4	32.0	33.0	66.9	67.4	67.2	67.3
	296	85.0	64.8	70.2	65.0	64.8	64.9	65.8	2.1	40.4	31.0	31.0	68.5	68.5	68.9	68.9
	300	90.4	65.1	72.5	64.7	64.7	64.8	66.2	—	—	36.0	35.0	72.4	72.4	74.4	74.0
One Wall Cold (East)	286	86.6	65.0	66.2	64.4	33.6	64.4	59.4	—	—	40.5	38.0	67.2	67.2	68.6	68.1
	297	84.8	64.9	65.2	64.2	34.1	64.2	59.2	1.2	40.75	40.5	37.5	65.2	65.3	66.3	66.0
3 Simulated Windows[d] 2 Simulated Outside Walls, Ceiling, North and West Walls Neutral	290	85.0	74.5	73.9	67.4	67.4	73.7	68.9	—	—	24.5	22.5	73.1	73.5	74.3	74.1
	291	84.6	73.7	72.4	67.1	67.1	72.1	68.0	2.1	39.9	29.0	27.0	68.3	70.0	70.0	70.0
Room Size 24½ ft x 12 ft x 12 ft high																
Uniform Environment	305	95.1	64.9	67.5	64.9	64.4	64.9	65.1	—	—	46.5	43.5	73.2	73.2	75.2	74.7
	306	85.0	64.2	66.5	64.4	64.5	64.4	64.7	—	—	29.5	28.0	70.1	70.1	71.6	71.2
	307	84.9	55.0	58.0	55.3	54.9	55.2	55.4	—	—	43.5	41.0	63.1	63.0	65.4	64.8
	308	85.9	73.7	75.6	74.4	74.1	74.4	74.3	—	—	17 0	16.0	77.5	77.2	78.2	78.0
	311	85.6	64.6	66.3	64.4	64.6	64.4	64.7	1.5	40.3	34.0	32.0	67.5	67.1	69.5	69.2
One Long, Cold Wall (East)	312	85.3	64.7	65.7	64.7	33.9	64.6	57.1	—	—	44.5	39.5	64.3	64.2	66.6	66.1
	313	84.9	64.9	64.6	64.3	33.7	64.1	56.8	1.5	21.5	49.5	44.5	58.7	60.4	63.1	62.8
3 Simulated Windows[d] 2 Simulated Outside Walls, Ceiling, North and West Walls Neutral	318	85.9	73.5	73.2	66.9	67.0	72.3	69.1	—	—	27.0	23.5	73.6	73.3	74.8	74.5
	319	85.0	71.4	69.5	67.0	67.2	69.3	67.4	1.5	20.9	34.0	31.0	65.8	67.2	69.4	69.3
	320	85.9	71.9	70.0	66.5	66.6	69.6	67.5	1.5	20.2	35.0	31.5	66.1	67.1	69.6	69.6
	321	86.6	74.6	74.3	67.0	67.2	73.2	69.6	—	—	26.0	24.0	73.7	73.6	75.3	75.0
Room Size 24½ ft x 12 ft x 4 ft high																
Uniform Environment	322	85.5	64.8	66.7	64.4	64.5	64.4	64.8	—	—	27.5	29.0	73.6	—	71.8	71.4
	323	95.3	64.8	66.7	64.3	64.5	64.4	64.8	—	—	42.5	44.0	77.5	—	75.1	74.6

[a] Area weighted average temperature of the ceiling and four walls.

[b] Determined from Figs. 1 and 4 of Reference 3 for the same AUST, panel temperature, infiltration temperature, and no. of air changes per hr. in *standard room.*

[c] Determined from Figs. 7 and 10 of Reference 3 for the same AUST, panel temperature, infiltration temperature, and no. of air changes per hr. in *standard room.*

[d] Simulated windows 4 ft x 5 ft with temperatures approximately 33 F.

non-uniform environment is 10.1 percent, and the maximum difference is 13 percent. In all cases, the observed outputs were higher than for the *standard room.*

As indicated for ceiling panels, room air temperatures in a floor panel heated room also appear to vary directly with changes in the ratio of heated to unheated areas resulting from different room sizes (see Fig. 2). Furthermore, convective heat transfer from the panel would be expected to increase with a decreased ratio of heated to unheated areas and cause a lower room air temperature and an increased air motion. These trends in panel output shown in Diagrams A and B of Fig. 2 are primarily due to the convective component, which is important in floor panel heating.

At first glance, the differences shown in Table 2 seem to be serious, but it should be noted that these differences would be appreciably reduced if, as in actual practice, comparisons were made on the basis of the same room air temperature rather than the same AUST and panel temperature. Since not all variables can be controlled independently, in a comparison on the basis of equal air temperature the panel temperature, panel output, and AUST must be adjusted.

The room heat balance must be satisfied. This is expressed by the formula:

$$C \times (\text{AUST} + t_o) \times (\text{unheated surface area}) = (\text{panel output}) \times (\text{panel area}) \quad \ldots (1)$$

where

C = average wall and ceiling conductance, Btu per (hr) (sq ft) (F deg inside surface to outside air)
t_o = outside air temperature, Fahrenheit

The relationship between room air temperature, AUST, and panel temperature must also be satisfied. This relationship (see Fig. 7, Reference 3) is given by the formula:

$$\text{room air temp} = 0.32 \times (\text{panel temp} - \text{AUST}) + \text{AUST} \quad \ldots (2)$$

For test No. 318, for instance, in which the difference between the observed and the calculated unit heat output was greatest, the comparison on the basis of equal room air temperatures would proceed as follows:

From test No. 318

Room Size	24½ × 12 × 12 ft
Room air temperature (60-in. level)	73.3 F
AUST	69.1 F
Unheated-surface area	1174 sq ft
Floor panel area	300 sq ft
Heat output per sq ft of panel area	27.0 Btu per hr

Average heat loss through the unheated surface area $= 300 \times 27/1174 = 6.9$ Btu per sq ft.

Any combination of conductance C (wall and ceiling) and temperature difference (AUST minus assumed outside temperature), the product of which would equal 6.9, could be thought of as simulating the unheated area of the Environment Laboratory during test No. 318. If an outside temperature of zero is assumed, the temperature difference is AUST—0 F or 69.1 deg. Then the C value for the walls and ceiling would be $6.9/69.1 = 0.1$ Btu per (hr) (sq ft) (F deg).

The problem now becomes one of finding the proper combination of AUST, panel temperature, and panel heat output which will maintain a room air tem-

perature of 73.3 F when the average conductance of the ceiling and walls is 0.10, and the outside temperature is 0 F. The room size is the same as that used in test No. 318. Equation 1 then becomes

$$0.1 \times (\text{AUST} - 0) \times (1174) = (\text{Panel heat output}) \times 300$$

and Equation 2 becomes

$$73.3 = 0.32 \times (\text{Panel temp} - \text{AUST}) + \text{AUST}$$

Since the two equations contain three unknowns, a trial and error method is indicated for their solution. Substituting corresponding values from Figs. 1 and 7 of Reference 3, the results shown in column 3 of Table 3 are found to satisfy both equations. For comparison, the actual observed values for test No. 318 are listed in column 2 and the calculated values from column 13 of Table 2 are given in column 4 of Table 3. It is evident from Table 3 that when the relatively slight corrections in AUST and panel temperature, necessary to maintain a 73.3 F room temperature, are made, good agreement in panel heat output is obtained. It may therefore be concluded that the heat flow and air temperature relationships developed in Reference 3 may be used without correction for designing panels for rooms of normal size and proportions. There is some evidence that the output of a floor panel increases slightly with ceiling height because of increased convection, and corrections may be necessary for ceilings exceeding the 12-ft height covered in the experiments.

EFFECT OF NON-UNIFORMITY OF PANEL TEMPERATURE ON PANEL PERFORMANCE

Test Methods: As described in a previous paper,[1] all of the interior surfaces of the Environment Laboratory consist of a number of metal sub-panels of various sizes which can be heated or cooled by liquid circulated through them. The $24\frac{1}{2} \times 12$ ft floor of the room is divided by a 6-in. wide $\times$ 12 ft long sub-panel into two 12 ft square areas, each covered by six 2×12 ft sub-panels laid with their long dimension parallel with the long dimension of the room. Each of these sub-panels is separately connected to the circulating system, and each

TABLE 3—COMPARISON OF CALCULATED PANEL PERFORMANCE USING STANDARD ROOM HEAT TRANSFER RELATIONSHIPS *vs.* OBSERVED PERFORMANCE

	Test No. 318	Basis of Comparison	
		Fixed Room Air Temp and Cond[a]	Fixed Panel Temp and AUST
	Observed	From Example	From Table 2
1	2	3	4
Air Temp (60-in. level)	73.3	73.3	74.5
Panel Output, Btu/(hr)(sq ft)	27.0	26.1	23.5
Panel Temp, F	85.9	86.2	85.9
AUST, F	69.1	67.2	69.1
Panel Temp minus AUST, F deg	16.8	19.0	16.8
C,[a] Btu/(hr)(sq ft)(F deg inside surface to outside air)	0.10	0.10	—

[a] Wall and ceiling conductance for zero F outdoor temperature.

connection is valved so that any sub-panel or combination of sub-panels may be heated or be shut off. The arrangement of sub-panels on the ceiling is the same as described above for the floor.

In the tests to be discussed, the valves to selected sub-panels were closed, so that the total heated area was less than the total area of the ceiling or floor surface. The valved-off sub-panels were neither heated nor cooled by liquid circulation, but reached an equilibrium temperature which resulted from the combined effects of radiative and convective heat exchange within the enclosure, the heat loss or gain from or to the back of the panels, and the heat conducted to the panels through piping and panel supports, and from adjacent heated panels. In all tests, the equilibrium temperature attained by these panels was high enough so that some heat was given off from them to the space.

Results reported here on this part of the program were made in the standard sized room. Similar results were obtained for rooms of two other sizes. Two general arrangements of heated panels were used. In some tests, the heated panel area was made up of *consecutively* located sub-panels 24 ft in length, starting at the east wall and extending as far across the floor or ceiling as desired. In other tests, alternate sub-panels were heated. In these tests, the first row of sub-panels along the east wall was heated; the second row was unheated, etc. In all cases, the 6 in. × 12 ft sub-panel was unheated.

Results with Non-Uniform Panel Temperatures: The results obtained in tests with only part of the ceiling or floor areas heated are tabulated in Table 4. Column 1 of Table 4 gives the percentage of the ceiling or floor area which was heated. The letters A or C appearing after the percentage values indicate *alternate* or *consecutive* arrangements of heated sub-panels. Column 3 indicates whether the environment was uniform or non-uniform. Columns 4 to 9 inclusive give the test conditions.

The observed panel heat outputs are tabulated in columns 10, 11 and 12 of Table 4. The values in column 10 are the heat outputs of the heated sub-panels at the observed temperatures listed in column 4. Similarly, column 11 gives the outputs of the unheated sub-panels at the temperatures in column 5. The values in column 12 are the area weighted average heat outputs of the entire ceiling or floor area, and the observed area weighted average surface temperatures are listed in column 6.

For comparison with the observed values in column 10, column 13 gives the calculated panel heat outputs which would have been obtained in the room with the same AUST and infiltration conditions which prevailed in the tests but with the *entire* ceiling or floor area heated to the temperature of the heated sub-panels shown in column 4. Similarly, the calculated heat outputs in column 14 are for the same AUST and infiltration conditions used in the tests but with the entire ceiling or floor area heated to the average temperature given in column 6. The values in columns 13 and 14 are based on data in References 2, 3 and 4.

Measured outputs from heated sub-panels given in column 10, Table 4, are higher in all tests than the corresponding calculated outputs shown in column 13. These greater outputs occurred for two reasons. First, and probably most important, when only a part of the ceiling was heated, the room air temperature was lower and the convection component of heat output was therefore increased. The second reason may be found in the fact that variations in ceiling tempera-

TABLE 4—OBSERVED PANEL OUTPUTS AND ROOM AIR TEMPERATURES FOR CEILING AND FLOOR PANELS AT NON-UNIFORM TEMPERATURES

Calculated Values for Panels at Uniform Temperature Shown for Comparison (Room Size 24½ × 12 × 8 Ft)

Area of Heated Panel Percent	Test No.	Temperature Environment	Observed Panel Temperature F			AUST	Infiltration		Panel Output Btu per(hr)(sq ft)					Room Air Temperature, F			
									Observed			Calculated		Experimental		Calculated[g]	
			Heated	Unheated[c,g]	Avg[d]		Changes per hour	Temperature F	Heated	Unh'ted	Avg[e]	Heated[f]	Avg[g]	30 in.	60 in.	30 in.	60 in.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Ceiling Panel																	
33C[a]	358	Uniform	119.9	72.4	87.9	64.3	none	—	64.8	4.4	24.2	61.2	23.6	65.6	66.7	65.2	66.2
49C	361		119.9	73.2	96.1	64.3	none	—	64.4	2.5	32.8	61.2	32.8	65.8	67.3	65.5	66.8
49A	344		119.9	79.5	99.3	64.8	none	—	65.7	8.1	36.4	60.6	36.0	66.5	68.4	66.2	67.6
49A	345		119.8	77.7	98.3	64.5	1	20.0	66.9	6.7	36.2	62.4	36.2	62.3	64.7	62.2	64.4
49A	346		119.1	76.8	97.5	64.5	2	20.0	66.9	6.4	36.0	62.4	35.6	57.8	61.4	58.5	61.3
49A	347		119.1	76.2	97.2	64.4	2¾	20.2	67.3	6.3	35.9	62.8	35.7	56.0	59.3	55.7	59.0
33C	359	Non-	119.3	67.5	84.5	56.7	none	—	73.2	3.5	26.3	67.1	26.5	58.2	60.9	57.9	59.5
49C	360	Uniform[b]	119.3	70.3	94.4	56.9	none	—	71.1	3.6	36.7	67.1	37.0	—	—	58.5	60.6
Floor Panel																	
33C	352	Uniform	85.4	66.8	72.9	64.7	none	—	30.8	2.0	11.4	28.8	10.8	67.2	67.3	67.5	67.4
49C	353		85.5	66.5	75.9	64.4	none	—	30 7	1.2	15.7	29.7	15.5	68.2	68.2	68.3	68.2
65C	354		85.0	66.9	78.7	64.3	none	—	29.8	0.8	19.7	29.2	19.5	69.2	69.2	69.3	69.0
49A	348		85.1	68.4	76.6	64.4	none	—	30.5	2.7	16.3	29.1	16.6	—	—	68.6	68.4
49A	349		85.8	67.1	76.3	64.9	1	20.7	36.8	2.3	18.8	33.3	18.8	65.7	66.0	66.2	66.2
49A	350		85.3	65.5	75.2	64.4	2	20.2	41.6	2.9	21.8	37.1	21.4	62.6	62.8	63.0	63.3
49A	351		85.3	64.5	74.7	64.4	2¾	20.8	43.6	3.1	23.0	40.3	23.6	60.3	61.1	61.0	61.4
33C	355	Non-	85.2	60.9	69.7	57.4	none	—	50.5	1.9	17.8	39.5	17.8	61.1	61.9	61.0	61.1
49C	356	Uniform	84.5	62.9	73.5	57.1	none	—	45.7	0.7	22.8	39.3	23.4	62.4	62.4	62.2	62.4
65C	357		85.3	63.5	77.7	57.9	none	—	43.6	−0.5	28.3	39.3	28.2	64.0	64.0	64.1	64.3

[a] C—Consecutive Panels; A—Alternate Panels Heated.
[b] One Long Wall 32 F; Other Unheated Surfaces 65 F.
[c] Neither Heated nor Cooled by Liquid Circulation—Neutral.
[d] Area Weighted Average of Columns 4 and 5.
[e] Area Weighted Average of Columns 10 and 11.
[f] For Test Conditions Given in Columns 3, 7, 8, 9 and Entire Panel Area at Temperature Given in Column 4.
[g] For Test Conditions Given in Columns 3, 7, 8, 9 and Entire Panel Area at Temperature Given in Column 6.

ture in themselves set up localized convection which tends to increase the convection output of the heated surfaces.

Certain trends may also be found by a comparison of the values in columns 10 and 13 of Table 4. It may be seen that the heat output per square foot of panel increases with a decrease in the percentage of the total floor or ceiling area that is heated. Heat outputs are also greater for alternately than for consecutively arranged heated areas, and for non-uniform than uniform environments. All of these increases are greater for floor panels than for ceiling panels. A more complete analysis of these involved relationships has been postponed until current work on the separation of the convective and radiative components of panel output have been completed. In contrast to the comparatively large differences between the values given in columns 10 and 13, the observed average heat outputs given in column 12 agree very well with the calculated average outputs in column 14. It is also significant that the observed air temperatures given in columns 15 and 16 are in excellent agreement with the calculated temperatures listed in the last two columns. It is therefore evident that the relationships between panel heat output and AUST, air temperature and panel surface temperature, as developed in References 2, 3 and 4 for panels of uniform temperature, also apply to average panel temperatures and average heat outputs when the panels are not at a uniform temperature.

This concept of treating the entire ceiling or floor as a unit may be advantageous. For example, suppose that conditions were such that the entire ceiling area at 100 F would give a total heat output equal to the room heat loss. On the assumption that the unheated part of the ceiling in the room under consideration is 70 F, any number of solutions would be possible as long as the area weighted average temperature of the entire ceiling is 100 F. If the heated portion of the ceiling were 120 F, the required panel area would be 60 percent of the ceiling area. If it were desirable to use 75 percent of the ceiling area for the heated panel, its surface temperature should be 110 F.

CONCLUSIONS

1. The effects of room size on the performance of floor and ceiling panels are relatively small so that the heat transfer relationships developed in the *standard room* and reported in References 2, 3 and 4, may safely be used without correction for the design of panel heating systems for any space of normal size and proportion.

2. Within the range of conditions tested, ceiling or floor panels comprised of heated and unheated sections have the same total heat output and produce the same room air temperature as if the entire area were heated to a uniform temperature equal to the area weighted average of the heated and unheated surfaces. The maximum test temperatures of the heated portions of floor and ceiling panels were 95 F and 140 F respectively.

CONTEMPLATED WORK ON PANEL HEATING

Work is currently in progress at the Laboratory to develop simplified procedures for the design of floor and ceiling panel heating systems. Concurrently, additional work is being done in the Environment Laboratory to evaluate separately the convective and radiative components of heat transfer from heating panels.

REFERENCES

1. A.S.H.V.E. RESEARCH REPORT No. 1444—The A.S.H.V.E. Environment Laboratory, by Cyril Tasker, C. M. Humphreys, G. V. Parmelee and L. F. Schutrum (A.S.H.V.E TRANSACTIONS, Vol. 58, p. 139).

2. A.S.H.V.E. RESEARCH REPORT No. 1473—Heat Exchanges in a Ceiling Panel Heated Room, by L. F. Schutrum, G. V. Parmelee and C. M. Humphreys (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 197).

3. A.S.H.V.E. RESEARCH REPORT No. 1490—Heat Exchanges in a Floor Panel Heated Room, by L. F. Schutrum, G. V. Parmelee and C. M. Humphreys (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 495).

4. A.S.H.V.E. RESEARCH REPORT No. 1499—Effects of Non-Uniformity and Furnishings on Panel Heating Performance, by L. F. Schutrum and C. M. Humphreys (A.S.H.V.E. TRANSACTIONS, Vol. 60, 1954, p. 121).

DISCUSSION

R. C. CHEWNING, Portland, Ore. (WRITTEN): A.S.H.V.E. research and the data provided by that research and reported in this and other papers are of great value to our Society. Several years ago, when the *boom* in panel heating started in this country, many engineers and contractors were faced with the problem of designing or installing their first panel heating system. Data were hard to obtain and largely unreliable. The research data being obtained by the Society are doing much to correct that situation.

Any criticism of the paper can be made only on the approach to obtaining the data as it affects its presentation. Much of the research performed by the Society is not known, used or appreciated. This paper offers an example of one of the reasons for this condition.

Reference is made to Table 1 in the paper. The tests run with a uniform environment show ceiling panel temperatures varying from 110.7 F to 120.3 F, panel outputs varying from 50.0 to 61.5 Btu per sq ft, and room temperatures varying from 62.0 F to 68.9 F at the 60-in. level. Considerable study is required to associate the variables in the various tests so that a knowledge of what the tests indicate may be obtained.

It would seem that curves or graphs properly designed would have illustrated the points being demonstrated much more clearly with less study being required of the reader.

In the tests on the non-uniform environment reported in the same table, the room temperatures are allowed to climb as high as 80 F at the 60-in. level. Much of the value of these data is lost because of the amount of study required to correlate the meanings of the figures. The many variables involved force the reader to accept the authors' conclusion without supporting evidence, unless he (the reader) undertakes a research project on the paper itself. Certainly it would have been possible to have maintained the same room temperature, the same AUST, and the same panel temperature in each test so that the Btu output per square foot could be observed to determine the effect of non-uniform environment. Panel temperature effects could have been demonstrated on a similar basis.

Very probably the method used was the easiest way to obtain the information desired, but the object of the Society is "to advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, *for the benefit of the general public.*" Certainly, unless the knowledge contained in research data is presented in a manner to promote its understanding by the members of the Society there is small chance of that knowledge being used for the *benefit of the public.*

R. J. MINDAK,* Chicago, Ill.: I would like to offer a constructive suggestion and also to reply to Mr. Chewning. From a quick perusal of this paper it is apparent that

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* Associate Engineer, Heat-Power Research Department, Armour Research Foundation.

there is a wealth of information contained therein—too much perhaps for one paper. In my opinion, the usefulness of the data would have been greatly enhanced if presented in two papers.

Commenting on Mr. Chewning's remarks, I can state that it is physically impossible to maintain all the temperatures constant during a series of tests. One of the three temperatures—panel surface, AUST, or room air—must be the dependent variable. I wonder if the problems involved in conducting these tests are fully appreciated by the previous critic. In collecting data such as these, one must compromise between ideal methods and practicability. This type of research requires considerable time and money and the investigator is faced with the problem of obtaining the best possible data in the shortest possible time at the least possible cost. Based on my own experiences in this same field of study I believe the data are excellent and were obtained in the best and quickest way possible.

R. G. VANDERWEIL, Boston, Mass.: I believe the data compiled and reported in this paper will ultimately help the practicing engineer because the results presented will facilitate simplified computation methods.

Is it possible now, to take one further step and to find one common solution applicable to any room of normal proportions which will state the number of Btu's delivered per square foot of floor (or ceiling) panel for a given panel surface temperature and a given design temperature? In other words, is the shape of rooms of normal practical proportions of small enough influence so that we can simplify the complicated calculations considering form factors? Such simplification was proposed in several engineering articles which I have written and which were published in 1945 and 1946.

M. L. GHAI,† Hartford, Conn.: First of all I would like to emphasize the importance of this paper. There are a number of applications involving rooms with high aspect ratios and one wonders if radiant heating could be used for these applications.

I am somewhat surprised at the conclusion of the authors that the room dimensions did not have any appreciable effect. This is probably due to rather low aspect ratios which were tried for experimentation.

In 1950 the writer worked on studies of radiant heating of spaces with large aspect ratios, such as railroad passenger car roomettes. Some of this work was published in the March, 1950, issue of *Railway Mechanical Engineering*. The tests indicated that the aspect ratios had considerable effect on the comfort. With radiant heat in the ceiling, stratification of air temperature was difficult to overcome, especially if the panels are designed for high output. The stratification of air temperature resulting from the non-uniformity of panel temperatures affected the panel performance.

It is also of interest to note the effect on variation of local radiant temperatures in the heated space resulting from non-uniformity of panel temperatures and room dimensions with high aspect ratio. Non-uniform panel temperatures can cause considerable variation of radiant temperatures, especially when accompanied by stratification of air temperatures, and can as a result give widely varying conditions of comfort.

The room sizes that the A.S.H.V.E. Laboratory experimented with are very well selected for residential heating. I wonder if it is possible to extend some of these tests to rooms of still larger aspect ratios. These would be of interest in radiant heating of railroad cars, buses, ships and probably airplanes.

Another project of research of general interest would be to study the effect of room size and non-uniformity of panel temperatures on the variation in the heated space of temperatures, and the variation of average air and radiant temperature indicating the comfort.

This is an excellent paper. I do not agree with the criticism that the paper does not present material in a manner in which it can be readily used by the application

† Director of Research and Development, Bush Manufacturing Co.

engineer. It often takes some time to understand and fully utilize the data presented in a good research paper of this type, but the importance of the paper should not be underemphasized.

AUTHORS' CLOSURE (L. F. Schutrum): Thank you, Dr. Ghai, for your interesting discussion of some of the problems encountered in panel heating. Studies of human comfort under panel heating and cooling conditions will be made in the Environment Laboratory when the panel cooling tests are completed. This work will be under the guidance of the TAC on Sensations of Comfort. It is not possible to increase the aspect ratio in the Environment Laboratory without installing partitions; however, this may be required in the future for the studies in human comfort.

The data in this paper do indicate that the shape of rooms of practical proportions, such as those found in normal residences, has small influence so that the simplification of otherwise complicated calculations is justified, as Mr. Vanderweil had pointed out a number of years ago.

Mr. Mindak's statement that this paper probably should have been two papers is shared by the authors.

In answer to a question by Mr. Chewning, it is not possible to maintain the same room air temperature, the same average unheated surface temperature, and same panel temperature in each test as suggested, for only two of these variables can be controlled. The panel output and one other variable, whether it is the air temperature, the panel temperature, or the AUST, are dependent variables. In the normal testing procedure, it was convenient to control the surface temperatures; consequently, the room air temperature and panel outputs were found by experimentation.

Mr. Chewning also criticized the manner in which the data are presented. It is not always possible to investigate the importance of the variables involved and maintain at the same time practical values for these variables. Mr. Chewning states that the many variables involved force the reader to accept the author's conclusion without supporting evidence. It is these many variables which make the problem complex and the papers involved. If a more simple method of presentation were known, it would have been used; however, it has always been the intent that sufficient fundamental data be included with each of these papers, so that the reader may draw his own conclusions.

The criticism that Laboratory papers are difficult to understand is not a new one and the Laboratory staff is continuously striving to make papers as interesting and readable as possible; however, this frequently presents a problem. Papers can be published as they are now in a detailed but unfortunately complex manner at the end of each phase of the overall research program, or can be held up until the final answers are known. In the latter method, work may go on for years without publication and consequently, the membership would not be familiar with what is being done. When the final paper is published, it would be limited in size by the requirements of the Society, and would probably give the results without sufficient evidence to support the conclusions.

The study presented in this series of papers has been guided by the Group B Subcommittee of the TAC on Panel Heating and Cooling. A similar series of experiments under the guidance of the Group A Subcommittee has resulted in information on plaster and concrete panels. At the present time, there is a subcommittee of these groups which is in the process of developing a design procedure. They are taking all of the work from these papers which is difficult to understand, and from these, working out a design procedure which we hope will be usable and reasonably correct. When the paper is published, you will have had the substantiating data, and it is hoped that you will find it satisfactory.

1517

No. 1517

COOLING LOADS FROM SUNLIT GLASS AND WALL

By C. O. Mackey* and N. R. Gay,** Ithaca, N. Y.

THE PURPOSE of this study was to add to the information concerning the relation between cooling load and instantaneous rate of heat gain by setting up on the hydraulic analogue a more complex problem than had previously been attempted. Cooling load was found at each hour of the day for a typical office of a multi-story office building under conditions where the sources of heat gain were assumed to be from sunlit glass and sunlit wall. These observed cooling loads may then be compared with the instantaneous rates of heat gain.

Room Size and Construction

The room size assumed was 20 ft by 20 ft by 8 ft. The room was assumed to have one exterior wall with 50 percent single-thickness window glass. The assumed construction of the exterior wall was 4 in. brick veneer on the weather side, 6 in. concrete, metal lath and plaster, furred. The office was assumed to be one of many in a multi-story office building and the floor and ceiling were of identical construction, 4 in. stone concrete. The interior walls or partitions were assumed to be of 3 in. hollow tile plastered on both sides.

The office to be studied was assumed to be adjacent on three sides to similar rooms so that the mid-plane of each partition is an adiabatic surface across which there is no flow of heat at any time. The ceiling of the office acts as the floor of a similar room above, and the floor of the office acts as the ceiling of a similar room below.

The properties of the materials used in the solution of this problem are shown in Table 1. It should be noted that the thermal conductances and heat storage capacities are expressed per square foot of floor.

Conditions of Heat Transfer

The exterior wall of the room was assumed to face south. There was assumed to be no infiltration of outdoor air and no internal sources of heat. Heat gain was entirely from glass and exterior wall transmission. The interior surfaces of the room were assumed to be in convection heat exchange with indoor air and in radiant energy exchange with each other. The solar and sky radiation directly transmitted by the glass were assumed to fall uniformly over the entire *floor;*

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** Associate Professor of Mechanical Engineering, Cornell University. Member of A.S.H.V.E.
Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Swampscott, Mass., June 1954.

TABLE 1—THERMAL CONDUCTANCES AND HEAT STORAGE CAPACITIES PER SQUARE FOOT OF FLOOR AREA

SURFACE	THICKNESS	AREA OF SURFACE PER SQ FT OF FLOOR	THERMAL CONDUCTANCE PER SQ FT OF FLOOR	HEAT STORAGE CAPACITY PER SQ FT OF FLOOR
	in.	ft²	Btu/(hr) (F deg)	Btu/F deg
Floor	2[a]	1	6.0	5.317
Ceiling	2[a]	1	6.0	5.317
Partitions	1.5[a]	1.2	1.464	2.4
Exterior Wall:				
Face Brick	4[b]	0.2	0.46	1.707
Concrete	6[b]	0.2	0.40	3.19
Lath and plaster, furred		0.2	0.20	0.21
Glass		0.2	1.79	0

[a] Thickness to mid-plane
[b] Total thickness

this assumption departs the most from actual conditions. The water vapor in the room air was assumed to absorb no radiant energy.

Solar and sky radiation directly transmitted by the single glass in the south wall was assumed to be representative of a location at 40 deg N latitude on August 1 with the values shown in Table 2.

For simplicity in establishing the convection and radiation heat transfer from the room-side surface of the glass, the temperature of the outdoor air in contact with the glass was assumed to be 83 F, the average daily design air temperature for clear atmospheres.

To establish the convection and radiation heat transfer from the room-side surface of the exterior wall, the sol-air temperature assumed for this wall was the design value for a surface with a solar absorptivity of 0.9 facing south; these values are given in Table 3.

TABLE 2—RADIATION DIRECTLY TRANSMITTED THROUGH SOUTH GLASS AND FALLING UNIFORMLY ON FLOOR

HOURS AFTER NOON	RADIATION TRANSMITTED PER SQ FT OF GLASS BTU/HR	RADIATION TRANSMITTED PER SQ FT OF FLOOR BTU/HR
0	98	19.6
1	90	18.0
2	69	13.8
3	42	8.4
4	18	3.6
5	11	2.2
6	7	1.4
7 through 17	0	0
18	7	1.4
19	11	2.2
20	18	3.6
21	42	8.4
22	69	13.8
23	90	18.0
Total for 24 hours	572	114.4

TABLE 3—ASSUMED VALUES OF THE SOL-AIR TEMPERATURE FOR THE SOUTH WALL

HOURS AFTER NOON	SOL-AIR TEMPERATURE, F	HOURS AFTER NOON	SOL-AIR TEMPERATURE, F
0	114	12	77
1	115	13	76
2	111	14	76
3	104	15	75
4	99	16	74
5	95	17	74
6	91	18	74
7	88	19	75
8	85	20	82
9	83	21	93
10	81	22	102
11	79	23	110

Average sol-air temperature for 24 hours, 89 F

It is necessary to decide upon various surface convection coefficients of heat transfer before the analogue is designed. The values assumed in this study are shown in Table 4.

For the exchange of energy by radiation between the room-side surfaces, it was assumed that the rate of such exchange was 1.0 Btu per (hr) (sq ft) (F deg). In other words, an equivalent convection coefficient of 1.0 was used. The angle factors, or the fraction of the radiant energy emitted by one surface that is incident upon another surface, were found and are given in Table 5.

THE ANALOGUE

The hydraulic analogue is shown in schematic form in Fig. 1 and as it actually appears in Fig. 2. Since the general principle of this analogue has been explained in a previous paper* this discussion is limited to an explanation of the specific

* Cooling Load From Sunlit Glass, by C. O. Mackey and N. R. Gay (A.S.H.V.E. TRANSACTIONS, Vol. 58, 1952, p. 321).

TABLE 4—ASSUMED VALUES OF CONVECTION COEFFICIENTS OF HEAT TRANSFER

SURFACE	CONVECTION COEFFICIENT OF HEAT TRANSFER BTU/(HR) (SQ FT) (F DEG)
Weather-side wall	4.0
Weather-side glass	4.0
Room-side wall	0.45
Room-side glass	0.45
Floor	0.60
Ceiling	0.30
Partitions	0.40

TABLE 5—ANGLE FACTORS FOR RADIATION EXCHANGES

SURFACES	ANGLE FACTOR	
	Symbol	Value
Floor to ceiling (Ceiling to floor)	F_{fc}	0.48
Floor to glass (Ceiling to glass)	F_{fg}	0.065
Floor to net wall (Ceiling to net wall)	F_{fw}	0.065
Floor to Partititions (Ceiling to partitions)	F_{fp}	0.39
Partitions to glass	F_{pg}	0.058
Partitions to net wall	F_{pw}	0.058

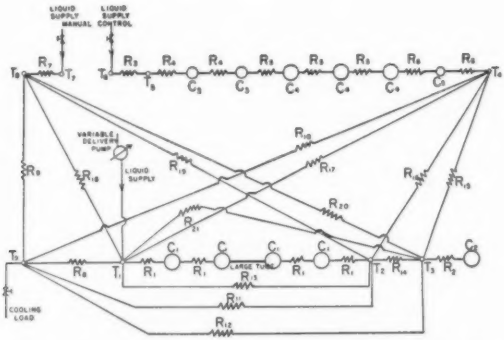


FIG. 1. SCHEMATIC DIAGRAM OF THE ANALOGUE

design for the particular study. Copper tubing of proper diameter and length was used for the equivalents of the thermal resistances R,† while glass tubes of proper areas of cross section were used for the equivalents of the heat storage capacities C. To indicate the surface temperatures, glass tubes T of the smallest diameter that would not introduce capillary effects were used. The resistance

† Letters and subscripts are defined in Tables 6, 7, and 8.

TABLE 6—THE RESISTANCE TUBES OF THE ANALOGUE (ALL RESISTANCES ARE FOR ONE SQUARE FOOT OF FLOOR AREA)

RESISTANCE TUBE SYMBOL	REPRESENTS	NUMERICAL VALUE OF THERMAL RESISTANCE (HR) (F DEG)/BTU
R_1	Conduction resistance of 1 in. of floor or ceiling	0.0833
R_2	Conduction resistance of one-half of partitions	0.683
R_3	Convection resistance of outdoor air near wall	1.25
R_4	Conduction resistance of 2 in. of face brick	1.09
R_5	Conduction resistance of 2 in. of concrete wall	0.833
R_6	Conduction resistance of one-half of furred lath and plaster	2.5
R_7	Combined resistances, outdoor air convection and glass conduction	1.81
R_8	Convection resistance of floor to room air	1.67
R_9	Convection resistance of glass to room air	11.1
R_{10}	Convection resistance of wall to room air	11.1
R_{11}	Convection resistance of ceiling to room air	3.33
R_{12}	Convection resistance of partitions to room air	2.08
R_{13}	Radiation resistance floor to ceiling	2.08
R_{14}	Radiation resistance ceiling to partitions	2.56
R_{15}	Radiation resistance partitions to wall	14.4
R_{16}	Radiation resistance ceiling to wall	15.4
R_{17}	Radiation resistance floor to wall	15.4
R_{18}	Radiation resistance floor to glass	15.4
R_{19}	Radiation resistance ceiling to glass	15.4
R_{20}	Radiation resistance partitions to glass	14.4
R_{21}	Radiation resistance floor to partitions	2.56

tubes are explained in Table 6, and the capacity tubes are explained in Table 7. The temperature tubes are explained in Table 8.

In the design of the analogue, the following scale factors were used:

Time: a time of one minute in the analogue represented a time of one hour in the prototype.

Temperature: a rise in level of the liquid in the tubes of one inch represented a temperature rise of 1 F deg in the prototype.

Rate of heat flow: a flow of liquid at the rate of 6 cc per min in the analogue represented heat flow at the rate of 1 Btu per (hr) (sq ft of floor) in the prototype.

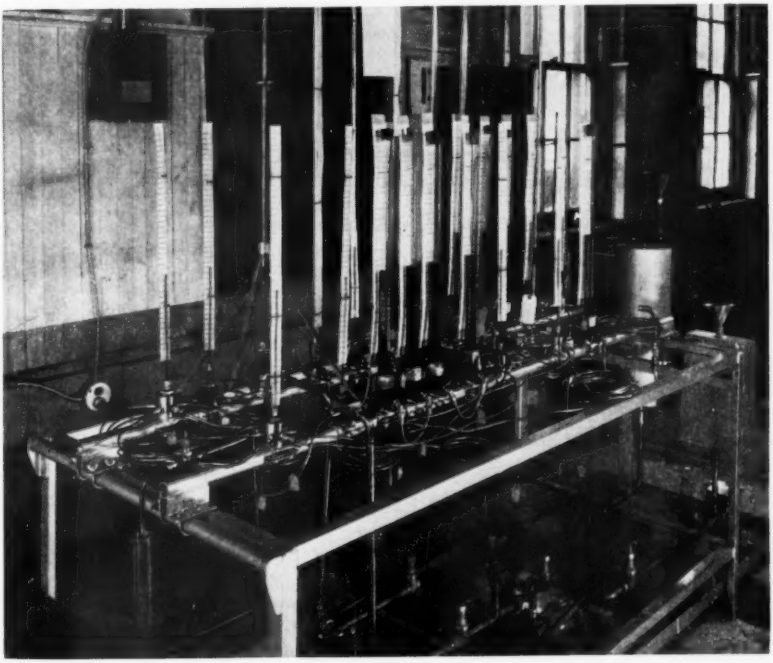


FIG. 2. THE HYDRAULIC ANALOGUE

A pump was designed to give a variable delivery with time; the complete cycle of the pump had a period of 24 min. With one design of cam, this pump reproduced the effect of a supply of heat directly to the floor at the rate shown in Table 2 for solar radiation transmitted directly through south glass. This liquid was supplied through a ¾ in. O.D. copper tube to temperature tube T_1.

Liquid was also supplied under manual control to temperature tube T_7; this liquid level was adjusted to represent the temperature of outdoor air near the glass. In these studies, this level was maintained constant to correspond to a steady temperature of 83 F.

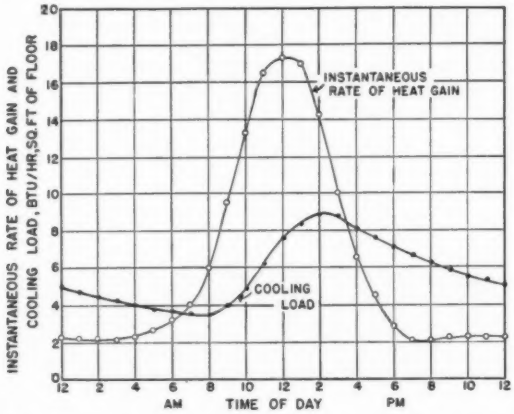


FIG. 3. CASE A: CONSTANT ROOM AIR TEMPERATURE

Liquid was supplied under manual control to temperature tube T_6; this liquid level was adjusted to represent the sol-air temperature for the exterior wall. For a south wall, the temperature schedule reproduced every 24 min was that shown in Table 3.

Liquid was drawn off through a needle valve in the line connected to temperature tube T_9. Any cycle of indoor air temperature may be maintained manually; for a constant temperature of the room air, for example, the level of the liquid in T_9 is held constant at the desired value. This liquid was collected in glass graduates, and the rate of its collection was measured. The cooling load may then be found either from the measured rate of collection of the liquid or by calculation from the observed instantaneous temperatures of the room surfaces, t_f, t_c, t_p, t_w, and t_g.

These studies are studies of periodic heat transfer in the unsteady state. The

TABLE 7—THE CAPACITY TUBES OF THE ANALOGUE (ALL CAPACITIES ARE FOR ONE SQUARE FOOT OF FLOOR AREA)

CAPACITY TUBE SYMBOL	REPRESENTS	NUMERICAL VALUE OF CAPACITY, BTU/F DEG
C_1	Capacity of 1 in. of floor or ceiling	2.658
C_2	Capacity of one-half of partitions	2.4
C_3	Capacity of 2 in. of face brick	0.953
C_4	Capacity of 2 in. of concrete wall	1.063
C_5	Capacity of metal lath and plaster	0.21

TABLE 8—THE TEMPERATURE TUBES OF THE ANALOGUE

TEMPERATURE TUBE SYMBOL	TEMPERATURE
T_1	Floor Surface
T_2	Ceiling Surface
T_3	Partition Surface
T_4	Room-side surface of wall
T_5	Weather-side surface of wall
T_6	Wall sol-air
T_7	Outdoor air near glass
T_8	Room-side surface of glass
T_9	Room air

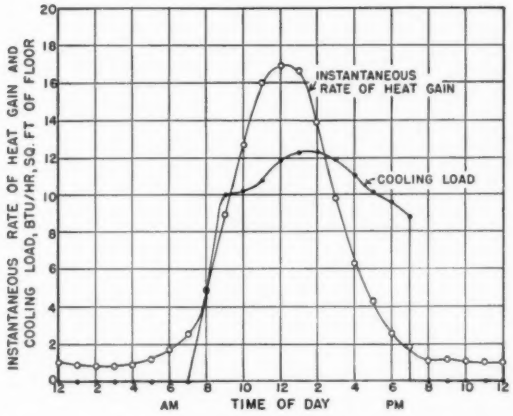


FIG. 4. CASE B: AIR CONDITIONING ON ONLY BETWEEN 7 A.M. AND 7 P.M.

period of the cycle was always 24 hr in the prototype (24 min in the analogue). In other words, the design conditions were assumed to recur each day for several days. In such a study, the starting conditions are not known; in other words, it is not known, for example, what the starting level of the liquid in temperature tube T_1 should be, since if it were known, a particular solution for the temperatures of the floor at some time of day would be known. It is necessary, then, to start with any liquid levels, repeat for several cycles the pump supply, the supply of liquid to tubes T_6 and T_7 and the removal of liquid from tube T_9.

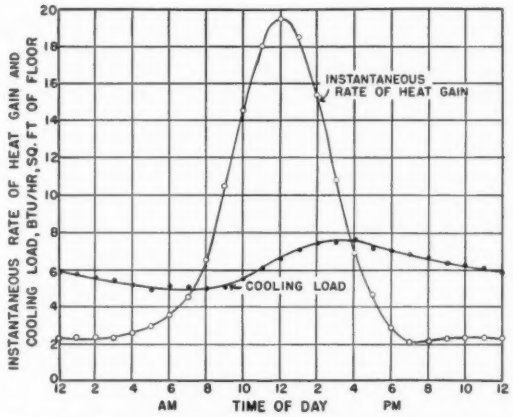


FIG. 5. CASE C: STEADY SUPPLY OF CONDITIONED AIR AT A FIXED DRY-BULB TEMPERATURE

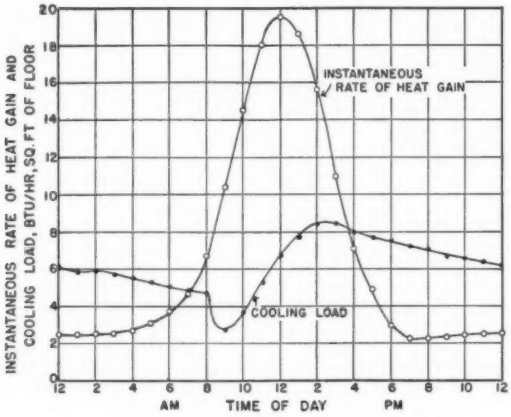


FIG. 6. CASE D: NIGHT PRE-COOLING

After several cycles, the effects of any wrong starting conditions will be eliminated. Then, further repetition will not appreciably change the levels of the liquid in any tube at a particular time in the cycle. When this *steady* periodic state has been reached, a solution to the problem set up on the analogue has been found.

SOLUTIONS

Case A: The temperature of the indoor air was assumed to be held constant for the 24-hr period at 75 F. The exposed wall and glass were assumed to face south. The sol-air temperature for the south wall, exclusive of glass, was that shown in Table 3, while the temperature of the outdoor air near the glass was assumed constant at 83 F. The pump delivery did not reproduce, exactly, the direct transmission of solar and sky radiation shown in Table 2, but it was close enough to these values for the purpose of this study. The results obtained with the analogue are shown in Table 9. The observed temperatures of the five different room surfaces appear in this table. The cooling load at each hour was calculated by finding the rate of heat transfer by convection from these surfaces to room air; a sample calculation is shown in Example 1.

The instantaneous rates of heat gain were also calculated from the observed temperatures. These gains consist of the solar and sky radiation directly transmitted by the glass, the transfer of heat by convection to room air from the inside surface of the glass and exposed wall, and the transfer of heat by radiation to surrounding surfaces from glass and exterior wall. A sample calculation is shown in Example 2.

The cooling loads and instantaneous rates of heat gain per square foot of floor are plotted against the time of day in Fig. 3.

EXAMPLE 1—CALCULATION OF COOLING LOAD

Time—12 noon: Cooling load per sq ft of floor is

$$0.6(81.7-75)+0.3(79.4-75)+0.4(480/400)\ (77.7-75)$$
$$+0.45(80/400)\ (79.3-75)+0.45(80/400)\ (81.4-75)=7.60 \text{ Btu/hr}$$

TABLE 9—RESULTS FOR CASE A (CONSTANT TEMPERATURE OF THE ROOM AIR OF 75 F)

SUN TIME	ROOM SURFACE TEMPERATURES, F					COOLING LOAD PER SQ FT OF FLOOR BTU/HR	INSTANTANEOUS RATE OF HEAT GAIN PER SQ FT OF FLOOR, BTU/HR			
								RADIATION & CONVECTION		
	FLOOR	CEILING	PARTITIONS	WALL	GLASS		DIRECT GLASS	GLASS	WALL	TOTAL
12 noon	81.7	79.4	77.7	79.3	81.4	7.60	16.10	0.94	0.33	17.37
1 p.m.	82.2	80.3	78.0	79.7	81.6	8.37	15.78	0.89	0.34	17.01
2	82.5	80.9	78.3	80.3	81.7	8.93	13.00	0.84	0.43	14.27
3	81.9	81.2	78.4	80.7	81.6	8.74	8.66	0.82	0.56	10.04
4	81.0	80.9	78.4	81.0	81.5	8.13	4.95	0.87	0.73	6.55
5	80.4	80.4	78.3	81.3	81.4	7.59	2.74	0.92	0.89	4.55
6	79.9	80.0	78.2	81.4	81.4	7.13	0.93	0.99	0.99	2.91
7	79.5	79.6	78.1	81.4	81.3	6.71	0.05	1.02	1.05	2.12
8	79.1	79.2	77.9	81.4	81.2	6.25	0	1.05	1.11	2.16
9	78.8	78.9	77.8	81.3	81.2	5.91	0	1.15	1.13	2.28
10	78.5	78.6	77.6	81.1	81.2	5.55	0	1.15	1.13	2.28
11	78.3	78.4	77.5	80.9	81.1	5.28	0	1.16	1.10	2.26
12 midnight	78.1	78.2	77.4	80.7	81.0	5.03	0	1.16	1.08	2.24
1 a.m.	77.9	78.0	77.3	80.5	81.0	4.78	0	1.19	1.05	2.24
2	77.7	77.8	77.2	80.2	81.0	4.52	0	1.23	1.00	2.23
3	77.6	77.6	77.1	79.9	81.0	4.33	0	1.25	0.93	2.18
4	77.4	77.4	76.9	79.6	80.9	4.02	0.18	1.27	0.89	2.34
5	77.3	77.3	76.8	79.3	80.9	3.85	0.56	1.29	0.82	2.67
6	77.2	77.2	76.8	79.1	80.9	3.74	1.19	1.30	0.78	3.27
7	77.1	77.1	76.7	78.8	80.9	3.58	2.04	1.32	0.71	4.07
8	77.1	77.0	76.6	78.6	80.9	3.48	3.98	1.33	0.67	5.98
9	77.9	77.2	76.7	78.5	81.0	4.07	7.72	1.29	0.56	9.57
10	78.9	77.6	76.9	78.5	81.1	4.90	11.63	1.21	0.46	13.30
11	80.4	78.4	77.2	78.8	81.2	6.22	15.00	1.07	0.38	16.45
Total for 24 hr						138.71	104.51	26.71	19.12	150.34

TABLE 10—RESULTS FOR CASE B (AIR CONDITIONING ON BETWEEN 7 A.M. AND 7 P.M. AND OFF FOR THE REST OF THE DAY)

SUN TIME	ROOM SURFACE TEMPERATURES, F					ROOM AIR TEMP., F	COOLING LOAD PER SQ FT FLOOR BTU/HR	INSTANTANEOUS RATE OF HEAT GAIN PER SQ FT OF FLOOR, BTU/HR			
								Direct Glass	Radiation & Convection		Total
	Floor	Ceiling	Partitions	Wall	Glass				Glass	Wall	
12 noon	84.9	82.5	80.2	81.0	82.1	75.0	11.87	16.10	0.56	0.25	16.91
1 p.m.	85.1	83.2	80.3	81.5	82.2	75.0	12.30	15.78	0.53	0.33	16.64
2	85.0	83.5	80.3	81.8	82.2	75.0	12.35	13.00	0.51	0.40	13.91
3	84.2	83.5	80.3	82.1	82.2	75.0	11.90	8.66	0.57	0.54	9.77
4	83.1	83.1	80.1	82.3	82.0	75.0	11.02	4.95	0.62	0.71	6.28
5	82.2	82.4	79.8	82.4	81.8	75.0	10.12	2.74	0.69	0.86	4.29
6	81.8	81.8	79.6	82.4	81.7	75.0	9.60	0.93	0.74	0.94	2.61
7	81.0	81.2	79.3	82.4	81.6	75.0	8.79	0.05	0.82	1.05	1.92
8	81.8	81.3	80.6	83.8	82.4	81.3	0	0	0.34	0.74	1.08
9	81.4	81.3	80.8	84.0	82.4	81.4	0	0	0.34	0.80	1.14
10	81.4	81.4	81.0	84.0	82.4	81.5	0	0	0.31	0.77	1.08
11	81.5	81.4	81.2	84.0	82.4	81.6	0	0	0.28	0.74	1.02
12 midnight	81.5	81.5	81.3	83.9	82.5	81.7	0	0	0.29	0.69	0.98
1 a.m.	81.6	81.5	81.4	83.8	82.5	81.7	0	0	0.27	0.65	0.92
2	81.6	81.6	81.5	83.6	82.5	81.8	0	0	0.25	0.57	0.82
3	81.7	81.6	81.6	83.5	82.6	81.8	0	0	0.27	0.53	0.80
4	81.7	81.7	81.7	83.3	82.6	81.8	0	0.18	0.25	0.45	0.88
5	81.8	81.7	81.8	83.1	82.6	81.9	0	0.56	0.23	0.38	1.17
6	81.8	81.8	81.8	82.9	82.6	81.9	0	1.19	0.22	0.31	1.72
7	81.8	81.8	81.8	82.7	82.6	82.0	0	2.04	0.21	0.24	2.49
8	81.7	81.6	81.2	81.9	82.3	78.5	4.80	3.98	0.50	0.39	4.87
9	82.3	81.4	80.3	81.0	81.8	75.0	9.99	7.72	0.71	0.48	8.91
10	82.8	81.5	80.0	80.7	81.8	75.0	10.16	11.63	0.69	0.37	12.69
11	83.8	81.9	80.0	80.8	82.0	75.0	10.80	15.00	0.66	0.31	15.97
Total for 24 hr							123.70	104.51	10.86	13.50	128.87

TABLE 11—RESULTS OF CASE C (STEADY SUPPLY OF CONDITIONED AIR AT A FIXED DRY-BULB TEMPERATURE)

SUN TIME	ROOM SURFACE TEMPERATURES, F					ROOM AIR TEMP., F	COOLING LOAD PER SQ FT FLOOR BTU/HR	INSTANTANEOUS RATE OF HEAT GAIN PER SQ FT OF FLOOR, BTU/HR			
								Direct Glass	Radiation & Convection		Total
	Floor	Ceiling	Partitions	Wall	Glass				Glass	Wall	
12 noon	81.3	78.9	77.2	78.9	81.4	75.2	6.62	18.20	1.02	0.30	19.52
1 p.m.	81.9	79.7	77.8	79.6	81.6	75.5	7.18	17.27	0.90	0.33	18.50
2	82.3	80.7	78.3	80.3	81.7	75.8	7.51	14.22	0.79	0.39	15.40
3	81.9	81.1	78.6	80.9	81.8	75.8	7.59	9.48	0.80	0.54	10.82
4	81.7	81.0	78.7	81.3	81.7	75.7	7.67	5.42	0.80	0.68	6.90
5	80.6	80.7	78.6	81.5	81.6	75.5	7.20	3.00	0.88	0.85	4.73
6	80.2	80.2	78.5	81.6	81.5	75.3	7.07	1.02	0.94	0.97	2.93
7	79.7	79.8	78.3	81.6	81.4	75.1	6.85	0.06	1.00	1.06	2.12
8	79.3	79.4	78.2	81.5	81.3	74.9	6.74	0	1.05	1.11	2.16
9	78.9	79.0	77.9	81.3	81.3	74.8	6.38	0	1.13	1.13	2.26
10	78.6	78.7	77.7	81.1	81.2	74.6	6.30	0	1.17	1.14	2.31
11	78.2	78.3	77.5	80.8	81.1	74.4	6.12	0	1.22	1.14	2.36
12 midnight	77.9	78.0	77.3	80.5	80.9	74.3	5.86	0	1.23	1.11	2.34
1 a.m.	77.7	77.8	77.0	80.2	80.9	74.2	5.85	0	1.29	1.08	2.37
2	77.4	77.5	76.8	79.8	80.9	74.0	5.57	0	1.35	1.04	2.39
3	77.2	77.3	76.7	79.5	80.8	73.9	5.47	0	1.37	0.99	2.36
4	76.9	77.0	76.4	79.2	80.8	73.8	5.19	0.20	1.44	0.97	2.61
5	76.7	76.8	76.3	78.8	80.7	73.8	4.91	0.61	1.44	0.89	2.94
6	76.7	76.7	76.2	78.5	80.6	73.6	5.11	1.30	1.44	0.84	3.58
7	76.6	76.6	76.0	78.2	80.6	73.5	5.05	2.24	1.48	0.78	4.50
8	76.6	76.4	75.8	77.8	80.6	73.4	5.01	4.36	1.52	0.70	6.58
9	77.4	76.5	75.9	77.7	80.7	73.7	5.10	8.46	1.45	0.58	10.49
10	78.4	77.1	76.2	77.9	80.8	74.0	5.58	12.74	1.33	0.49	14.56
11	79.8	77.8	76.7	78.3	81.0	74.5	6.15	16.42	1.17	0.39	17.98
Total for 24 hr							148.08	115.00	28.21	19.50	162.71

TABLE 12—RESULTS FOR CASE D (NIGHT PRE-COOLING)

SUN TIME	ROOM SURFACE TEMPERATURES, F					ROOM AIR TEMP., F	COOLING LOAD PER SQ FT FLOOR BTU/HR	INSTANTANEOUS RATE OF HEAT GAIN PER SQ FT OF FLOOR, BTU/HR			
								Direct Glass	Radiation & Convection		Total
	Floor	Ceiling	Partitions	Wall	Glass				Glass	Wall	
12 noon	81.1	78.8	77.1	78.8	81.4	75.0	6.73	18.20	1.07	0.30	19.57
1 p.m.	81.7	79.8	77.6	79.5	81.5	75.0	7.70	17.27	0.96	0.37	18.60
2	82.1	80.5	78.0	80.2	81.7	75.0	8.42	14.22	0.91	0.48	15.61
3	81.7	80.9	78.3	80.7	81.7	75.0	8.49	9.48	0.89	0.60	10.97
4	80.9	80.8	78.3	81.0	81.6	75.0	7.99	5.42	0.92	0.75	7.09
5	80.4	80.4	78.2	81.3	81.5	74.9	7.70	3.00	0.97	0.91	4.88
6	79.8	80.0	78.1	81.3	81.3	74.7	7.47	1.02	1.00	0.93	2.95
7	79.3	79.5	77.9	81.3	81.3	74.5	7.23	0.06	1.10	1.10	2.26
8	78.9	79.0	77.7	81.2	81.2	74.3	7.04	0	1.16	1.16	2.32
9	78.5	78.6	77.5	81.0	81.0	74.2	6.70	0	1.18	1.17	2.35
10	78.1	78.3	77.3	80.8	80.9	74.0	6.56	0	1.22	1.19	2.41
11	77.8	77.9	77.0	80.5	80.0	73.8	6.40	0	1.28	1.19	2.47
12 midnight	77.5	77.6	76.8	80.2	80.8	73.7	6.16	0	1.34	1.17	2.51
1 a.m.	77.2	77.3	76.5	79.8	80.8	73.6	5.87	0	1.41	1.12	2.53
2	76.9	77.1	76.3	79.4	80.7	73.3	5.96	0	1.45	1.08	2.53
3	76.7	76.8	76.1	79.0	80.6	73.2	5.76	0	1.48	1.04	2.52
4	76.4	76.5	75.9	78.7	80.6	73.1	5.52	0.20	1.54	0.99	2.73
5	76.2	76.3	75.7	78.3	80.5	73.0	5.36	0.61	1.56	0.92	3.09
6	76.0	76.1	75.5	77.9	80.5	73.0	5.05	1.30	1.60	0.85	3.75
7	75.9	75.9	75.4	77.6	80.4	73.0	4.84	2.24	1.60	0.79	4.63
8	76.0	75.7	75.3	77.4	80.4	73.0	4.77	4.36	1.61	0.74	6.71
9	77.1	76.1	75.8	77.7	80.8	75.0	2.74	8.46	1.41	0.52	10.39
10	78.1	76.8	76.1	77.9	80.9	75.0	3.72	12.74	1.31	0.44	14.49
11	79.8	77.7	76.6	78.3	81.2	75.0	5.32	16.42	1.20	0.36	17.98
Total for 24 hr							149.50	115.00	30.17	20.17	165.34

EXAMPLE 2—CALCULATION OF INSTANTANEOUS RATES OF HEAT GAIN

Time—12:00 noon: Direct transmission by glass is 16.1 Btu per hr for each sq ft of floor; this figure comes from the calibrated pump discharge and is not exactly the same as the values given in Table 2.

Convection from glass	$=0.45(80/400)\ (81.4-75)$	=	0.576	
Radiation from glass to floor	$=0.065(81.4-81.7)$	=	−0.020	
Radiation from glass to ceiling	$=0.065(81.4-79.4)$	=	0.130	
Radiation from glass to partitions	$=0.058(1.2)(81.4-77.7)$	=	0.257	
Convection and radiation from glass per sq ft floor	$=0.576+0.367$	=	0.94	Btu/hr
Convection from wall	$=0.45(80/400)(79.3-75)$	=	0.387	
Radiation from wall to floor	$=0.065(79.3-81.7)$	=	−0.156	
Radiation from wall to ceiling	$=0.065(79.3-79.4)$	=	−0.007	
Radiation from wall to partitions	$=0.058(1.2)(79.3-77.7)$	=	0.111	
Convection and radiation from wall per sq ft floor	$=0.387-0.052$	=	0.33	Btu/hr
Total instantaneous rate of heat gain per sq ft floor	$=16.10+0.94+0.33$	=	17.37	Btu/hr

The total cooling load in 24 hr should check the total instantaneous rate of heat gain for the same period. Actually, these totals differ by about 8 percent, which is a reasonable accuracy for such an analogue study.

It will be observed that the maximum cooling load is about 9.0 Btu per hr for each sq ft of floor at about 2:00 p.m.; the maximum instantaneous rate of heat gain is about 17.4 Btu per hr at 12:00 noon. *In this case, then, the peak cooling load is 52 percent of the peak instantaneous rate of heat gain and occurs with a time lag of 2 hr.* The ratio of the peak cooling load to the 24 hr average instantaneous rate of heat gain is 1.44.

Case B: The structure and its orientation were assumed to be the same as in *Case A.* There was assumed to be a supply of conditioned air for 12 hr of the day between 7 a.m. and 7 p.m. to maintain a constant temperature of the indoor air of 75 F; during the remainder of the day, there was no drawing off of liquid from the analogue to simulate a shutting down of the air conditioning equipment. The results obtained with the analogue are shown in Table 10 and in Fig. 4.

Case C: The structure and its orientation were assumed to be the same as in *Case A.* A constant setting was used on the needle valve to draw off liquid from the system. In effect, this is similar to a constant rate of supply of conditioned air at a constant dry-bulb temperature to the office during the entire 24-hr period; the cooling load remains nearly constant. The results obtained with the analogue are shown in Table 11 and in Fig. 5.

Case D: The structure and its orientation were assumed to be the same as in *Case A.* The only difference was that the indoor air temperature was assumed to be constant at 75 F between 8 a.m. and 5 p.m., while this temperature was allowed to drop to as low as 73 F during the evening hours. The results obtained with the analogue in this test are shown in Table 12 and in Fig. 6.

CONCLUSION

For the several different methods of operating an air conditioning system to handle the cooling loads simulated for *Cases A* to *D*, it will be noted in Figs. 3 to 6 that in each case the peak cooling load is only a fraction of the peak instantaneous rate of heat gain.

DISCUSSION

G. V. PARMELEE, Cleveland, Ohio: I think our profession owes a debt of gratitude to Professors Mackey and Gay for continuing the work in this very important field which, I hope you realize, is only a beginning. It is a very difficult problem to handle and it is gratifying to me to see that the fluid analogue is capable of representing quite satisfactorily a fairly complex real system.

I have several questions I would like to ask the authors. One of these concerns the capacity elements used in the fluid analogue. I wonder if they have investigated the effect of varying the number of capacity tubes in the floor and ceiling circuits. I don't think it is as important in the exterior wall, where there are already a large number of tubes, and because no radiation from the sun falls on the inner surface of this wall. I do think this is important with respect to those room surfaces on which solar radiation does fall.

Second, it would be of great interest to know how long it took the system to reach the steady periodic state. In many locations we don't often have the same daily temperature and radiation cycles day after day. Consequently I think it would be important to know something about the transient heat flow which they undoubtedly have observed and for which they may have data.

In this particular system two or three components have been combined depending on how one looks at it. One is the direct radiation transmitted through the glass. That contributes most of the load. Second is the transfer from the glass to the space by virtue of the difference between the glass temperature and the space temperature. That can be a fairly sizeable component if the glass area is large. The third is the heat flow through the wall itself, which is not too large.

I would like to see the separate contributions to the cooling load of each of these three components. It might show that separate factors should be applied to the instantaneous rates of heat gain due to each of these components. If this were so, results such as these could be more readily applied to instantaneous gain calculations as we now make them.

If each of the load components, or input functions, were treated separately, I think it would show that a small part of the solar radiation which entered the room never becomes a part of the load. This is because the indoor surfaces of the room are warmed by this radiant energy. This reduces the inflow of heat through the wall and through the window due to the weather conditions on the other side and therefore represents a loss of some of the solar energy which entered the room.

Finally, I would like to know if Professor Mackey could estimate what would happen to the peak cooling load if the air temperature in contact with the outer surface of the glass was allowed to vary throughout the day. I recognize that this is an added experimental difficulty.

R. W. MCKINLEY, Pittsburgh, Pa.: I would like to ask the authors whether or not the analogue could be conveniently set up to help two groups in the Society which are interested in a related aspect of the heat flow through glass problem: the TAC on Heat Flow Through Glass and the Joint A.S.H.V.E. and *I.E.S.* Committee on Loads Related to Lighting.

It occurred to me as I read through the paper and listened to the presentation that it might be possible to establish a uniform lighting level in this space produced by a combination of solar illumination and electric illumination and then determine the relative contribution to the cooling load from each source.

I believe there is a good possibility of establishing interconnected operating controls over fenestration and electric lighting such as to minimize the total lighting load on the cooling system. Because this load can be a very large one in well-lighted buildings, any means of reducing it seems worthy of careful study.

C. F. KAYAN, New York, N. Y.: I would like to add a few words of commendation to the authors for developing this concept of computational operation, from first treating the problem mathematically to then ending the thermal circuit, and finally employing a technique to solve the thermal circuit.

We recognize there are various ways in which we could solve this thermal circuit. One way is the hydraulic method. Another way is an electrical simulating method. Both ultimately will have their place and I dare say eventually we may use mechanized computational procedures to handle this. Certainly we are now going forward. As one who has been concerned with a method of analogical computation I applaud these results so graphically shown this morning. I would like to commend Professors Mackey and Gay for what they have done, and particularly also to commend their co-worker, "Professor" Leopold of Philadelphia, who as an electrical engineer, mind you, worked with the *hydraulic* analogue and often chided me as a mechanical engineer, for working with that "horrible" *electrical* analogue.

R. G. VANDERWEIL, Boston, Mass.: I should like to express a few words of gratitude, after listening to this paper. I still remember how a good many years ago I listened to Professor Kayan's description of his electrical analogger. I was then impressed by the simplicity of his method compared with that of analytical computation, and after his presentation discussed with him my hope that he would apply his method to the panel heat flow problem.

Now in applying the *fluid type* analogue, a further step has been taken, resulting not only in simplicity of solution but also in vivid and visual description of the problem. As a mechanical engineer, I join Professor Kayan in appreciating the results obtained. For the first time in my practice I could *see* a temperature, I could *see* heat storage, and for its visibility I believe that further development of this method, and its application to problems of a more complex nature, should be vigorously supported.

It was striking to note how a new approach to an old problem could help to explain it more clearly and to understand it more readily. Many times I have wondered just what might be the influence of heat storage on a computed heat gain. With this method we can not only simply allow for storage, but can see it.

I should like to ask two questions: First—the authors mentioned a west cam on the pump. Why do you change the cam with exposure? Second—I wondered whether in their method of computing tube diameters for those tubes representing storage, the viscosity of the fluid was considered. I assume there is a relationship between viscosity, tube diameter, and time?

C. S. LEOPOLD, Philadelphia, Pa.: When I was first faced with the problem of calculating heat transfer due to high temperature radiation impinging on a solid, and in particular a panel cooling system, the only thing I was sure of was that the mathematical tools at my disposal were not adequate.

The hydraulic analogue is considerably older than my paper* on the subject. The major contribution of my paper was to solve heat transfer problems in terms of introducing given quantities of energy within a system and then determining the temperature distribution. In previous work it was customary to impress the assumed temperatures on the system and determine the distribution of energy.

A hydraulic analogue was selected in order to see the process and in that particular phase I think it is superior to the electrical analogue. The electrical analogue provides results with far greater rapidity and for repetitive work it is certainly to be preferred.

Care must be exercised in not generalizing data of this type. Prof. Mackey carefully pointed out that he assumed for his problem that all radiation impinged on

* Hydraulic Analogue for the Solution of Problems of Thermal Storage, Radiation, Convection and Conduction, by Charles S. Leopold (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 389).

the floor. If there were a white venetian blind partially closed, a relatively small part of the energy would strike the floor and a larger portion would strike the ceiling. The results would be considerably different.

In a previous paper† the data obtained from a model of a room and an artificial sun showed that although the energy gain from the sun varied over a wide range, very little difference in ceiling temperature was required to compensate for the condition in which there was a white venetian blind with slats at 45 deg and no shade. This result was explained by the fact that for no shade and a 3-hr exposure, there was a large thermal storage in the floor whereas with the venetian blind there was no appreciable storage in the floor.

AUTHORS' CLOSURE (C. O. Mackey): I wish to thank those who have contributed to this paper by their discussion.

First, in reply to questions raised by Mr. Parmelee. In regard to the number of capacity elements, those of you who are familiar with the mathematics of the solution will realize that the use of the analogue with lumped resistances and capacities is like the Schmidt method of calculation in which the material is divided into slabs—the greater the number of slabs, the greater the accuracy. In this analogue, floor and ceiling slabs were 1 in. thick. From previous study of a simpler case for which an exact solution was known, we decided that the use of smaller slab thickness was not demanded for estimating heat flow with an accuracy of 5 percent. Mr. Parmelee also raised a question about the number of repeated cycles required to attain the steady periodic state. It generally took about four cycles. We have some data which show what was happening during this transient state period. Mr. Parmelee's remarks were well taken and I should emphasize again that the end results shown in this study would only be realized if the conditions of the design day were repeated for several days. From the tabular data it is possible to compute the contribution of each room surface to the cooling load at a particular time, but it is not possible to answer all of Mr. Parmelee's questions. For example, in *Case A* at 2 p.m., the contributions of the several room surfaces to the peak cooling load are as follows: floor, 50.5 percent; ceiling, 19.8 percent; partition, 17.7 percent; glass, 6.7 percent; wall, 5.3 percent. The effect of a variable air temperature on the weather side of the glass was not deemed sufficiently important to include as a further complication.

In deference to Professor Kayan and others who have commented on the use of the electrical analogue in solving similar problems, we do not regard the fluid analogue as a superior tool. This problem may be solved with calculating machines; it may be solved by an electrical analogue; it may be solved by a fluid analogue. The electrical analogue has some advantages over the fluid analogue, but for one purpose the fluid analogue has no equal. If you want to *see* what happens in a complex problem in unsteady state heat transfer, the fluid analogue has no superior because the tubes are giant thermometers. You have often thought—"if I could only have a thermometer here to see what is going on". You *see* in the fluid analogue what it is more difficult to visualize from the mathematics or from the electrical analogue.

The remarks of Mr. Vanderweil accent the importance of the visual representation provided by the fluid analogue. The cam of the variable-delivery pump is changed with the orientation of the glass. The pump consists of two bellows operated by a phasing mechanism. When these bellows are in phase, the pump gives maximum delivery; when the bellows are completely out of phase, there is no delivery. The cam adjusts the phasing mechanism so that the rate of delivery changes with time to simulate the rate at which solar energy passes through glass of a particular orientation. The viscosity of the model liquid has no effect upon sizing capacity tubes. If a 1 in. rise in liquid level represents a 1 deg rise in temperature, if 6 cc of liquid flow per minute represents heat flow at the rate of 1 Btu/hr, if 1 min in the model

† The Mechanism of Heat Transfer, Panel Cooling, Heat Storage: Part II—Solar Radiation, by Charles S. Leopold (*Refrigerating Engineering*, June 1948).

TABLE A—RESULTS FOR ROOM WITH WEST-FACING GLASS

(Room Air Temperature 75 F)

SUN TIME	ROOM SURFACE TEMPERATURES, F					COOLING LOAD PER SQ FT OF FLOOR BTU/HR	INSTANTANEOUS RATE OF HEAT GAIN PER SQ FT OF FLOOR, BTU/HR			
								RADIATION & CONVECTION		
	Floor	Ceiling	Partitions	Wall	Glass		Direct Glass	Glass	Wall	Total
12 noon	78.5	77.8	77.3	78.9	81.2	4.95	3.80	1.23	0.56	5.59
1 p.m.	80.5	78.3	77.6	79.2	81.3	6.49	11.60	1.07	0.46	13.13
2	83.6	79.6	78.3	79.9	81.7	9.17	25.40	0.85	0.33	26.58
3	87.0	81.7	79.4	81.0	82.1	12.50	36.00	0.53	0.21	36.74
4	89.3	84.1	80.5	82.4	82.6	15.30	41.00	0.29	0.23	41.52
5	89.7	86.1	81.5	83.9	82.9	16.78	39.00	0.15	0.45	39.60
6	87.5	87.0	81.9	84.8	82.9	16.00	26.20	0.22	0.77	27.19
7	85.4	86.0	81.6	85.1	82.5	14.29	1.20	0.32	1.07	2.59
8	84.3	84.7	81.3	85.4	82.3	13.11	0.00	0.44	1.34	1.78
9	83.1	83.5	80.7	85.2	82.2	11.71	0.00	0.61	1.48	2.09
10	82.4	82.7	80.5	85.1	82.0	10.93	0.00	0.66	1.56	2.22
11	81.5	81.9	80.1	84.7	81.8	9.90	0.00	0.74	1.59	2.33
12 midnight	80.8	81.2	79.8	84.1	81.6	9.06	0.00	0.80	1.52	2.32
1 a.m.	80.3	80.6	79.3	83.6	81.6	8.29	0.00	0.90	1.48	2.38
2	79.9	80.1	79.0	83.0	81.4	7.69	0.00	0.92	1.39	2.31
3	79.4	79.6	78.7	82.4	81.4	7.04	0.00	1.01	1.30	2.31
4	79.0	79.2	78.5	81.9	81.3	6.53	0.00	1.04	1.22	2.26
5	78.6	78.8	78.1	81.4	81.3	5.93	0.00	1.06	1.16	2.22
6	78.4	78.5	77.9	80.8	81.3	5.57	1.20	1.17	1.03	3.40
7	78.3	78.3	77.8	80.4	81.2	5.36	2.00	1.17	0.94	4.11
8	78.2	78.1	77.6	80.0	81.1	5.10	2.40	1.17	0.86	4.43
9	78.0	77.9	77.5	79.6	81.1	4.83	2.80	1.23	0.77	4.80
10	78.0	77.8	77.3	79.3	81.1	4.68	3.20	1.17	0.71	5.08
11	78.1	77.7	77.3	79.0	81.1	4.68	3.40	1.23	0.62	5.25
Total for 24 hours						225.89	199.20	19.98	23.05	242.23

represents 1 hr in the prototype, a storage volume of 12 cc per 1 in. rise in liquid level represents a material that stores 2 Btu per deg rise in temperature. The viscosity of the liquid affects the length of the tube of a particular diameter that must be used to simulate the thermal resistance.

Mr. McKinley raised the question of use of the fluid analogue by two other committees of the Society. How best to solve problems in heat transfer in the unsteady state of particular interest to air conditioning engineers is a question for additional study. The A.S.H.V.E. Research Laboratory may need some general purpose analogue or analogues which might be either fluid analogues, electrical analogues or both.

The movie showed results for west-facing glass while this paper gives results for south-facing glass. Table A gives results obtained with the fluid analogue for west-facing glass and a constant room air temperature of 75 F for a room of the same construction and in the same location as the room of the paper.

With west orientation of the exposure, the maximum cooling load is about 40 percent of the maximum instantaneous rate of heat gain; for south-facing glass, the corresponding ratio was found to be 52 percent. The difference is in the expected direction, because the more peaked, or the greater the amplitude, of the instantaneous rate of heat gain, the more effective is the structure as an energy-storage device.

1518

No. 1518

COOLING STUDIES IN A RESEARCH HOME

By W. S. HARRIS* AND P. J. WAIBLER,** URBANA, ILL.

ON JANUARY 2, 1940, the *Institute of Boiler and Radiator Manufacturers* entered into an agreement with the University of Illinois Engineering Experiment Station for cooperative research in the field of steam and hot water heating. This agreement has been continued since that date. During the first year the *I=B=R* Research Home was built and fully instrumented to serve as a laboratory in which the performance of heating systems could be observed with laboratory precision while operating under conditions typical of actual usage. Since construction, the Research Home has been continuously used for that purpose.

The Institute is represented by a Research Committee which proposes problems for investigation that are of the greatest interest to the manufacturers and installers of steam and hot water heating systems. This committee unanimously recommended that during the summer of 1953 an investigation be started having as its long range objective the determination of the most practical means of providing summer comfort in homes having a steam or hot water heating system. This paper discusses results obtained during the first summer of operation.

EQUIPMENT

Research Home: The *I=B=R* Research Home, Fig. 1, is a two-story building, typical of the small, well-built American home. The construction is brick veneer on wood frame, and all of the outside walls and the second story ceiling are insulated with mineral wool batts 3⅝ in. thick. A vapor barrier was placed between the studs and the plaster base to retard the passage of water vapor from the rooms into the insulation during the winter months. The calculated coefficient of heat transmission, *U*, for the wall section is 0.074 Btu per (hr) (sq ft) (F deg). All windows and outside doors are weatherstripped. The windows are double hung wood sash with the exception of those in the kitchen which are of wood casement construction.

The floor plans of the Research Home are shown in Fig. 2, and a summary of the estimated cooling loads is given in Table 1. An outdoor temperature of 96 F and an indoor temperature of 75 F were selected as design conditions. No estimate was made of the probable latent load. Heat gains through the walls and roof were estimated using the equivalent temperature differential method developed by Stewart and described in Chapter 13, Cooling Load, HEATING VENTILATING AIR CONDITIONING GUIDE 1953. The heat gains through the windows were also estimated using the method recommended in THE GUIDE. Infiltration loads were estimated by using the air change method and assuming the same number of air changes for summer operation as is recommended in

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THE GUIDE for winter heat loss calculations. Undoubtedly these infiltration values are high for summer operation, but even though the assumed infiltration rates were high the heat gains due to infiltration are extremely small in comparison to the other loads (see Table 2).

In estimating the design cooling load no allowances were made for such internal loads as lights and occupancy; however, during the tests the house was occupied by an average of four people during the day and two at night. There was a normal usage of lights, but there was no cooking in the house during the testing season.

Cooling System: The cooling system was designed for use in conjunction with a hot water heating system. A schematic diagram of the system, consisting of



FIG. 1. *I=B=R* RESEARCH HOME

a water chiller and room units resembling convectors through which either chilled or hot water can be circulated, is shown in Fig. 3. As indicated the piping was so arranged that water was circulated through the chiller and the room units during summer operation, and during the winter the same piping could be used to circulate water between the units and the boiler. In sizing the piping, the required water circulating rates for both summer and winter must be considered and the system designed to handle whichever is the larger. In this particular case it was found that a piping system sized for winter heating was adequate for summer cooling as well. All piping was insulated with a vapor proof insulation to prevent condensation during the summer operation.

Fig. 4 shows a cross section through one of the room units illustrating the arrangement of fan, filter and coils. Condensate from the coils was collected by the drip pan from which it could be conveyed to a drain by suitable piping as shown in Fig. 3. During the tests the condensate from each unit was collected in a container and measured so as to determine the latent load being carried by each room unit.

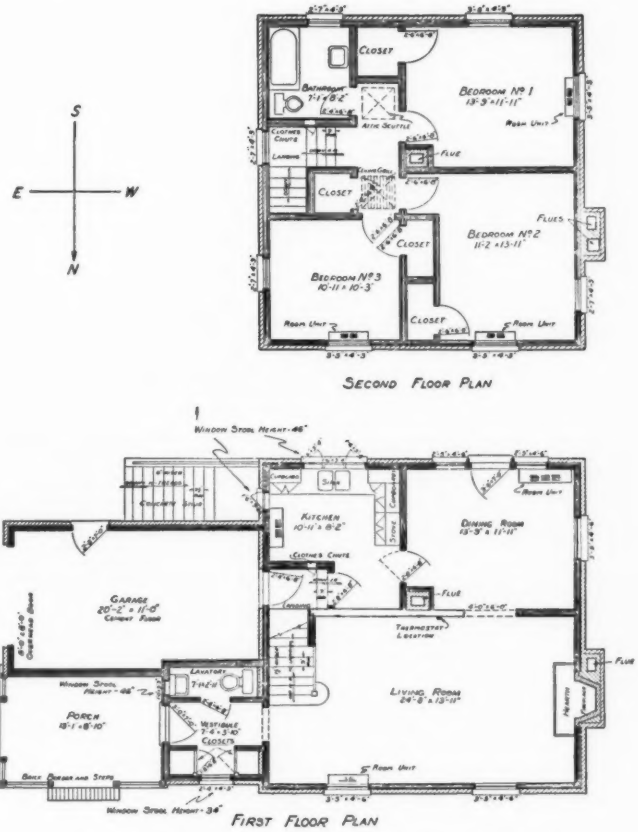


FIG. 2. FLOOR PLAN OF *I=B=R* RESEARCH HOME

All of the units with the exception of the one in the dining room were of the same size and selected from those commercially available. The measured air delivery for each unit at both high and low fan speeds is given in Table 1. The units were representative of available capacities most nearly approaching the estimated cooling loads of the rooms in which they were used. The manufacturer's rating for each unit, as well as the design sensible cooling load for each room, are given in Table 1.

The water chiller was rated at two tons capacity. The compressor was directly connected to a 2 hp, 220 volt, single phase motor. Both the motor and the condenser were water cooled.

Controls: The fans on the room units were manually controlled. They were operated continuously during tests when the windows were closed. During the

series of tests conducted with the windows open at night, the fans were shut off for the open window period.

The operation of the circulating pump and the compressor motor was controlled by a room thermostat located on an inside wall (see Fig. 2) in the living room 30 in. above the floor. When the air temperature rose above the thermostat temperature setting, the thermostat started both the circulating pump and the compressor which operated until the air temperature in the living room dropped below the thermostat temperature setting. The operating differential of the thermostat was approximately 2 deg. The chiller was protected by a limit control which would stop the compressor motor at any time the water temperature in the chiller dropped below approximately 39 F.

Testing Apparatus in Research Home: Approximately 100 copper constantan thermocouples made of No. 22 B and S gage wire were permanently installed in the walls and ceilings in order to measure temperatures at important points in

TABLE 1—DATA ON HOUSE AND ROOM UNITS

	Design Sensible Cooling Loads[a]	Rated Capacity of Installed Room Unit			Measured Air Delivery of Room Unit Installed	
			Sensible Cooling[b]		Fan Speed	
		Water Flow Rate	High Fan Speed	Low Fan Speed	High	Low
Room	Btu/hr	GPM	Btu/hr	Btu/hr	CFM	CFM
Kitch.....	1,942	0.3		2,300	200	151
Din.....	5,240	1.2	5,700		317	281
Liv.....	1,138	0.3		2,300	192	167
NE Bed.....	1,207	0.2		1,660	208	159
NW Bed.....	1,383	0.2		1,660	231	185
SW Bed.....	2,562	0.5		2,670	217	171
Total.....	13,472	2.7	16,290			

[a] Based on maximum outdoor temperature = 96 F; indoor temperature = 75 F
[b] Based on 45 F entering water temperature; 75 F entering air temperature

the structure under various operating conditions. At each of eight locations, one on each exposure of each story, nine thermocouples were installed to provide the data for complete temperature gradients through the walls. Another group of approximately 50 thermocouples provided for the measurement of air temperatures at various levels in the center of each room, in the attic, and in the basement. A third group made it possible to study the performance of the component parts of the cooling system. Provision was made for measuring the temperature of the water entering and leaving each room unit as well as the temperature of the water entering and leaving the chiller.

A central switchboard was located in the basement, and all the thermocouples were connected to selector switches on this board. In this way the electromotive force of each thermocouple could be read quickly on a precision potentiometer used in connection with a highly sensitive galvanometer. A 10 point recording potentiometer used in connection with an auxiliary switchboard made

TABLE 2—COMPARISON OF CALCULATED AND OBSERVED COOLING LOADS MAXIMUM OUTDOOR DRY BULB TEMPERATURE = 100 F AVERAGE INDOOR DRY BULB TEMPERATURE = 75 F; SIX-ROOM UNITS IN OPERATION SEPTEMBER 1, 1953

ROOM	CALCULATED LOADS AT 1:00 P.M. (MAX TOTAL LOAD) BTU PER HR						OBSERVED MAX LOAD (1:30 TO 4:00 P.M. DST)			CALCULATED SENSIBLE MINUS OBSERVED TOTAL LOAD	OBSERVED ROOM-AIR TEMP AT 30 IN. LEVEL	
	Walls	Ceiling	Glass	Infiltration	Total Walls and Ceiling	Sensible Total	Sensible	Latent	Total		Max	Min
Kitchen	241	—	1,735	131	241	2,107	1,718	0	1,718	389	77	74
Din Rm	173	—	4,925	604	173	5,702	2,360	27	2,387	3,315	77	73
Liv Rm	276	14	990	277	290	1,557	2,134	171	2,305	—748	76	74
Total 1st Story	690	14	7,650	1,012	704	9,366	6,212	198	6,410	2,956		
N.E. Bed	245	159	908	140	404	1,452	1,741	74	1,815	−363	75	72
N.W. Bed	131	222	890	140	353	1,383	2,193	56	2,249	−866	74	71
S.W. Bed	240	220	1,962	140	460	2,562	2,142	59	2,201	361	76	73
Total 2nd Story	616	601	3,760	420	1,217	5,397	6,076	189	6,265	−868		
House Total	1,306	615	11,410	1,432	1,921	14,763	12,288	387	12,675	2,088		

it possible to obtain either instantaneous readings or continuous printed records of the emf produced by the thermocouples in any selected group.

The rate of flow of water through the chiller was made by installing an elbow meter† in the return main. It was used in connection with a sensitive differential pressure recorder. This type of meter introduces no additional resistance to the flow of water.

The pressure drop through each room unit and the connecting piping was measured by a mercury manometer. Prior to the testing season, the relationship between the pressure loss and rate of water flow was determined for each of the room units so that the pressure losses could be used to measure the respective water flow rates during a test.

Recording thermometers were used to make continuous records of the air temperatures in each of the six rooms. The moisture content of the air was measured by means of four humidity indicators, one recording hygrometer, and one wet- and dry-bulb recorder which were checked periodically with an aspirated psychrometer. The electrical inputs to circulator and compressor motors were measured by means of integrating watt-hour meters having scale divisions

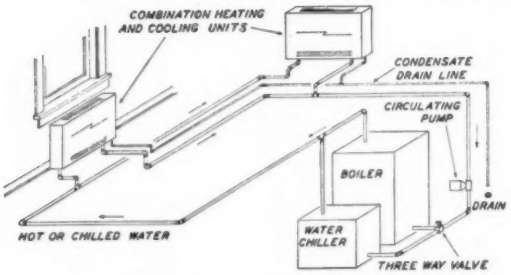


FIG. 3. HEATING AND COOLING SYSTEM

of 10 watt-hours. Self-starting electric clocks were wired into the compressor and circulator motor circuits to indicate the total time of operation.

TEST CONDITIONS AND METHODS

The following three test conditions were used:

1. Windows closed at all times and no ventilation air other than normal infiltration supplied to the house.
2. Windows closed at all times but mechanical ventilation supplied by the continuous operation of a kitchen exhaust fan, located in the east window of the kitchen.
3. Windows open from 10:00 p.m. to 7:00 a.m. In this series the cooling equipment was not operated during the time the windows were open. For part of the tests in this series no ventilation air was supplied during the time the windows were closed; in the remaining tests the kitchen ventilating fan was used.

In some tests, the number of room units in operation was varied from a maximum of six, one in each of the major rooms, to a minimum of two, one in the living room and the other in the northeast bedroom.

The selection of these test conditions and methods of operation made possible

† The Use of an Elbow in a Pipe Line for Determining the Rate of Flow in the Pipe (*University of Illinois Engineering Experiment Station Bulletin No.* 289, 1936).

three separate comparisons of operating costs and comfort produced in the house: (1) the effect of the number of room units in operation; (2) the effect of ventilation air; and (3) the effect of night air cooling.

In all tests the thermostat was set to maintain an average indoor temperature of 75 F. Except for some special tests, each test was 24 hr in length. In addition to those conditions continuously recorded by instruments, it was customary to read all room-air temperatures at levels of 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling, the relative humidity in each room,

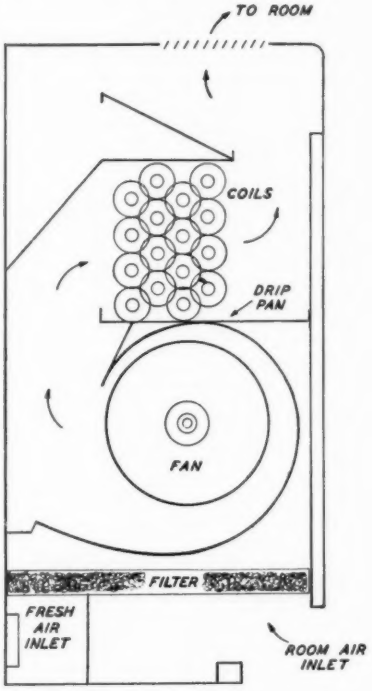


FIG. 4. CROSS SECTION THROUGH ROOM UNIT

and the quantity of water removed from the room air two to four times during each test. The power consumption of the compressor and circulating pump motors was recorded for each test period. The temperature of the water entering and leaving the chiller and each of the room units as well as the respective flow rates were also taken during at least one cycle of operation for each of the tests.

RESULTS

Operation on a Typical Day: Fig. 5 is a graphic log of test conditions for two representative 24-hr test days which began at 7:00 p.m. The solid lines of the

plot represent the data of a test period in which the chiller operation was controlled at all times by the room thermostat. Three room units were in use; one in the living room, one in the northeast bedroom, and one in the south bedroom. House air was exhausted by means of a continuously operating kitchen fan. All windows were kept closed.

The outdoor temperature was 85.5 F at 7:00 p.m., fell through the night to a low of 67 F at 7:00 a.m. and rose again to 96 F at noon with approximately seven hours when the temperature exceeded 90 F. The 24 hr average temperature was 82.3 F. The outdoor wet bulb temperature had a minimum of

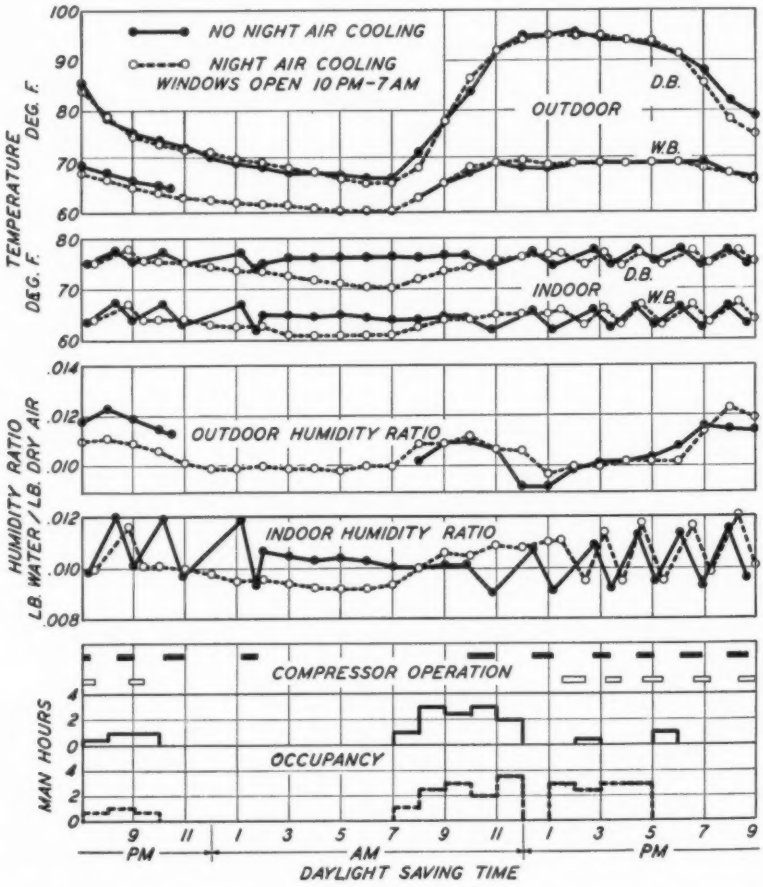


FIG. 5. TEMPERATURE AND HUMIDITY WITH AND WITHOUT NIGHT AIR COOLING

60.5 F at 7:00 a.m. and a maximum of 69.5 F at noon. The relative humidity (not shown) increased to a maximum of 72 percent during the night and decreased to a minimum of 26 percent during the warmest part of the day.

The chiller was operating at the beginning of the test period, 7:00 a.m., and operated four times during the night, the final operation ending at 1:45 a.m. An 8-hr period followed with no compressor operation, the first operation during daylight hours occurring at 9:55 a.m. The chiller operated at approximately 2 hr intervals for the rest of the test period.

Each operation of the chiller and circulating pump caused a drop of about 2 deg in the average indoor dry bulb temperature and a drop of about 4 deg in the average indoor wet bulb temperature. The average indoor relative humidity for the 24 hr was about 55 percent.

Comparison of Measured and Calculated Cooling Loads: Table 2 shows a comparison of calculated and observed cooling loads for a day when the maximum outdoor temperature was 100 F. Calculating the cooling loads by the procedure outlined in THE GUIDE indicated that the maximum load should occur about 1:00 p.m. DST. The observed maximum load did not occur until about 1:30 p.m. and continued until about 4:00 p.m. The total measured maximum sensible cooling load for the house was only 12,288 Btu per hr as compared to a calculated maximum sensible load of 14,763 Btu per hr.

The calculated heat gains for rooms having southern exposures were high compared to the actual measured cooling load, while for the rooms having northern exposures the calculated loads were lower than those actually observed. When operating with all room doors open there is air movement from room to room, and hence there is some transfer of load from one room to another. The measured loads (Table 2) were obtained after the water flow to each of the room units had been adjusted to give the best possible balance of room air temperatures. There was only 3 deg difference in temperature between the warmest and the coolest rooms in the house, the rooms with south exposures being warmer. Thus, it is probable that any transfer of load from one room to another was from those on the south side to the north side. This would make the differences between measured and calculated loads even greater than those indicated by Table 2.

The magnitude of the differences between measured and calculated heat gains were sufficiently large that they could not be attributed to errors in the estimation of wall and ceiling gains alone. In fact, the data seem to indicate that most of the discrepancy between calculated and measured loads must be in the estimated heat gains through the glass areas. When estimating cooling loads, it is common practice to assume that all radiant energy transmitted through glass is immediately available to heat the air in the rooms. Actually this energy is not transformed to heat until it strikes some solid object, which is warmed and then heats the room air by convection. There is a finite time lag between the time the radiant energy is transmitted through the windows and the time it actually warms the room air. Furthermore, the convective heat transfer rate between the objects warmed by radiation and the air in the room is not necessarily as high as the rate at which solar radiation is received by objects in the room. Ignoring these facts would tend to make the maximum estimated instantaneous load occur earlier and have a larger magnitude than the actual instantaneous loads on the cooling equipment.

The present methods of estimating solar heat gains through glass areas are really methods of determining the instantaneous transmittance of solar energy through the glass and do not necessarily reflect the rate at which the solar energy eventually warms the air in the room. The latter affects the load on the cooling equipment.

Relationship Between the Number of Units and Cooling Performance: One series of tests was made in which the windows were closed at all times, and the number of room units in operation was changed from a maximum of six to a minimum of two. From this series three days having approximately the same average outdoor temperatures have been selected. During one of these days the Research Home was cooled with all six room units operating. During the second day four of the room units were in operation, and during the third day only two room units were used.

Table 3 presents a summary of outdoor temperatures and general operating conditions for these three days. With all six room units in operation the average indoor air temperature was maintained at 76 F, but due to short operating time and the high water temperature only 8.4 lb of water were removed from the air. As a result, the relative humidity was almost 70 percent, too high for comfort. The operating cost was $1.14 per day for electricity and water used. Between an on- and off-period, the variation in air temperature in any room was 3 to 4 deg, with a similar variation in average air temperature from room to room.

When the number of units in operation was reduced to four (Test II) the capacity of the system was correspondingly reduced which made it necessary to operate for a longer period of time and with a lower water temperature in order to cool the room air to 75 F. At these operating conditions, more than 23 lb of water were removed from the air, resulting in a relative humidity of approximately 63 percent. Eliminating the use of two of the six room units reduced the combined cooling capacity of the room units by about one third. The total operating cost during Test II was $1.00 as compared to $1.14 for Test I. Practically all of the reduction in operating cost was attributed to the fact that only four room unit fans were in operation during Test II whereas six were in operation during Test I.

In Test III the room unit in the northeast bedroom did not have sufficient capacity to cool the entire second story. As a result, second story air temperatures were 2 or 3 deg higher than normal, and the average indoor temperature for the test was 78.9 F, as compared to approximately 76 F for Tests I and II. Again, approximately 24 lb of water were removed from the room air and the resulting indoor relative humidity was just under 60 percent. Since no changes were made in the water chiller, it had a greater capacity than required by the two room units, and therefore the compressor motor was cycled by action of the limit control during each cooling cycle. The operating cost for Test No. III was only $0.82. About 90 percent of the difference in the operating cost of Tests I and III was the result of using only two unit fans in Test III, while 6 were used in Test I. The other 10 percent was due to an actual reduction in the total time of operation of the compressor in Test III.

In the course of the summer of testing, the flow rate of the condenser cooling water was varied in an attempt to find the minimum acceptable. In the analysis of data of comparable days, as in Table 3, it was found that the actual condenser

cooling water rates were not the same. To reduce cost comparisons to a comparable basis, actual cooling water consumptions were corrected to a common flow rate. It was assumed that the changes in cooling water flow rate involved in these would not change the chiller performance appreciably.

This series of tests demonstrated that as long as room doors are not closed a satisfactory cooling job can be done without providing a unit in every room.

TABLE 3—EFFECT OF NUMBER OF UNITS ON PERFORMANCE

TEST I—July 28, 1953

Maximum Outdoor Temperature	92.5 F
Average Outdoor Temperature	83.1 F
Average Outdoor Humidity Ratio	0.0155 lb/lb
Average Indoor Temperature	76.3 F
Average Indoor Relative Humidity	69.9 percent
Average Indoor Humidity Ratio	0.0135 lb/lb
Total Water Removed From Air	8.4 lb
Compressor Operating Time	7.52 hr
Pump Operating Time	7.52 hr
Total Operating Cost (per day)	$1.14
Average Temperature of Chilled Water Supplied to Units	54.3 F

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed
Unit in Operation	Yes	Yes	Yes	Yes	Yes	Yes
Fan Speed	High	High	Low	High	High	High
Max Air Temp, 30 in. Level, F.	75.5	77.0	76.5	79.5	79.5	79.5
Min Air Temp, 30 in. Level, F.	71.0	73.5	74.5	76.5	76.0	76.0
Water Removed from Air, lb.	0.7	0.8	2.9	1.2	1.8	1.0

TEST II—July 30, 1953

Maximum Outdoor Temperature	90.0 F
Average Outdoor Temperature	81.6 F
Average Outdoor Humidity Ratio	0.0175 lb/lb
Average Indoor Temperature	75.4 F
Average Indoor Relative Humidity	63.3 percent
Average Indoor Humidity Ratio	0.0119 lb/lb
Total Water Removed From Air	23.4 lb
Compressor Operating Time	8.10 hr
Pump Operating Time	8.10 hr
Total Operating Cost (per day)	$1.00
Average Temperature of Chilled Water Supplied to Units	48.6 F

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed
Unit in Operation	No	No	Yes	Yes	Yes	Yes
Fan Speed	—	—	High	Low	Low	Low
Max Air Temp, 30 in. Level, F.	78.0	77.5	76.5	76.5	76.0	76.0
Min Air Temp, 30 in. Level, F.	76.5	76.0	74.5	73.0	72.0	72.5
Water Removed from Air, lb.	—	—	9.2	5.9	4.8	3.5

TEST III—August 3, 1953

Maximum Outdoor Temperature	95.5 F
Average Outdoor Temperature	81.3 F
Average Outdoor Humidity Ratio	0.0168 lb/lb
Average Indoor Temperature	78.9 F
Average Indoor Relative Humidity	59.5 percent
Average Indoor Humidity Ratio	0.0127 lb/lb
Total Water Removed From Air	24.4 lb
Compressor Operating Time	7.03 hr
Pump Operating Time	10.37 hr
Total Operating Cost (per day)	$0.82
Average Temperature of Chilled Water Supplied to Units	42.4 F

Room	Kitch	Din Rm	Liv Rm	SW Bed	NW Bed	NE Bed
Unit in Operation	No	No	Yes	No	No	Yes
Fan Speed	—	—	High	—	—	Low
Max Air Temp, 30 in. Level, F.	78.5	78.5	77.0	81.0	81.0	79.0
Min Air Temp, 30 in. Level, F.	77.5	77.5	75.0	80.5	80.0	76.5
Water Removed from Air, lb.	—	—	14.3	—	—	10.1

The room air temperatures reported in Table 3 show that in Tests II and III the average air temperature in the living room was only 2 deg lower than that in the dining room or kitchen even though no cooling was provided in the latter two rooms. Also, the cyclic variation in the first story temperatures between the on- and off-periods was less in Tests II and III where only the unit in the living room was in operation than in Test I where cooling units were in operation in all three rooms. Test III shows that except for somewhat higher average air temperatures, exactly these same characteristics were exhibited on the second story even though these rooms were more or less isolated from one another. If the unit in the northeast bedroom had sufficient capacity to reduce the average air temperature in this room 1.5 deg lower than it did in Test III, there is every reason to believe that the same reduction in temperature would have occurred in the other bedrooms. Therefore, two units of proper capacity could have cooled all of the Research Home with a maximum room-to-room temperature difference of about 2 deg, while the operating cost would be much less than if the units were used in all six rooms.

It may be concluded from this series of tests that decreasing the number of room units in operation resulted in:

1. Decrease of the cyclic air temperature variation in the rooms.
2. Slight improvement in the temperature balance between rooms up to the point where the number of units in operation was not adequate to carry the entire load.
3. Increase in the amount of water removed from the air and a lower relative humidity in the rooms.
4. Reduced operating cost.

These tests demonstrate that with the cooling equipment used in these tests, the installation of room units having a capacity in excess of that required decreases the quality of performance from the standpoint of comfort and increases the cost of operation. On the other hand, it is quite essential that the capacity of the water chiller be sufficient to reduce the temperature of the water in the system to the minimum value in a reasonable period of time. This is important if adequate dehumidification is to be obtained.

Night Air Cooling: The curves in Fig. 5 show the indoor and outdoor conditions for two 24-hr test periods. These two test periods were almost exact duplicates as far as outdoor temperature and humidity were concerned. During one of the test periods the windows were closed all of the time and the cooling system was controlled by the thermostat. During the other test period all windows were opened from 10:00 p.m. to 7:00 a.m. The cooling unit was not operated during the time the windows were open.

While the outdoor temperature was 95 F from noon to about 5:00 p.m., it was 75 F or less from 9:00 p.m. to 8:30 a.m., reaching a low of approximately 67 F between 5:00 and 7:00 a.m. A diurnal temperature variation of about 25 deg is typical of summer conditions in Urbana. Even though the summer of 1953 was one of the warmest on record in Urbana, there were only five days during the summer in which the minimum outdoor temperature was above 75 F and 16 days when the minimum outdoor temperature was between 70 and 75 F.

Fig. 5 shows that during the night, both the indoor dry bulb temperature and humidity ratio were lower for the test period in which the windows were opened than for the test made with closed windows. During the rest of the day the indoor conditions were essentially the same for both tests.

The estimated sensible cooling load between the hours of 9:00 a.m. and 9:00 p.m. DST and the measured total load during each operating cycle of the two tests are shown in Fig. 6. It can be seen that the measured loads were well under the calculated load during the heat of the day. For the test made with closed windows, the measured load exceeded the calculated load after 8:00 p.m. Because of heat storage in the house and furnishings there was some cooling load all night long during the test with closed windows even though the outdoor temperature was well below the desired indoor temperature. On the other hand, there was no need of cooling in the house from 10:00 p.m. until 1:30 the following afternoon when operating with open windows at night.

Opening the windows at night had little or no effect on the maximum cooling load, and hence on the capacity of cooling equipment required; it merely short-

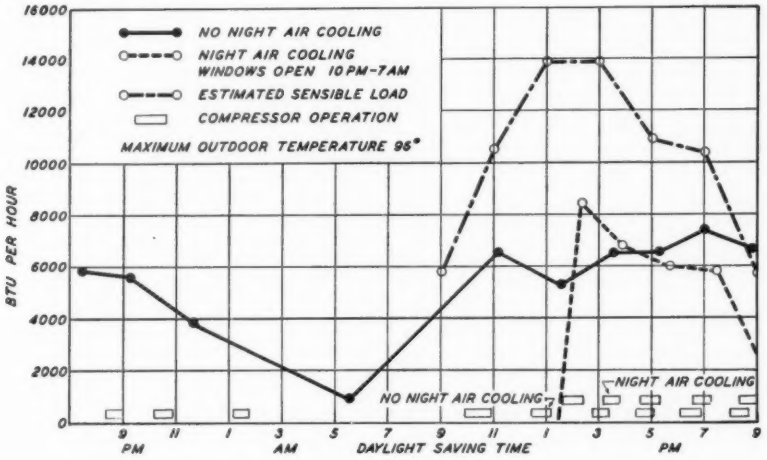


FIG. 6. COOLING LOAD WITH AND WITHOUT NIGHT AIR COOLING

ened the length of time the cooling system had to operate in order to maintain comfortable indoor temperatures. In the section of this report in which costs are discussed, it is shown that by making use of night air cooling operating costs were reduced by almost 50 percent.

Ventilation Air: To demonstrate the effect of ventilation air on the cooling load a few tests were made in which the cooling system was operated with closed windows at all times, and air was exhausted by means of an exhaust fan located in the east window of the kitchen and operated continuously during the test period. No special provision was made for the entrance of outdoor air into the house. The measured free air delivery of the exhaust fan was 430 cfm.

The tests revealed that the ventilation air increased the maximum load by approximately 25 percent and the total daily load by about 10 percent. At no time did those working or sleeping in the house notice that the operation of the exhaust fan increased the quality or *freshness* of the air in the house. Although there seemed to be no need of mechanical ventilation, it is probable that in

homes it might be required at times when food is being cooked, unusual occupancy or heavy smoking. Ventilation during these times could be provided by a small kitchen exhaust fan. If windows are opened at night, there is even less need of mechanical ventilation.

Operating Costs: The operating costs are determined by the quantity of cooling water used, the power consumption of the compressor, the pump and the fans on the room units; and the respective power and water rates. Table 4 shows a break-down of the power and water consumptions and operating costs based on average rates at Urbana. Four methods of operation are represented, and in each case the maximum outdoor temperature was approximately 95 F.

TABLE 4—OPERATING COSTS

TEST	A		B		C		D		E	
No. of Units Operating	6		3		3		3		3	
Night Air Cooling	No		No		No		No		Yes	
Ventilation Air	No		No		Yes		Yes		Yes	
Maximum Outdoor DB	92.5		95.0		96.0		96.0		96.0	
Minimum Outdoor DB	71.5		74.0		72.0		67.0		67.0	
Average Outdoor DB	83.1		84.4		82.3		80.3		81.0	
Average Indoor DB	76.3		77.0		76.0		76.0		74.7	
	KWH	Cost	KWH	Cost	KWH	Cost	KWH	Cost	KWH	Cost
Compressor	16.3	$0.407	13.6	$0.340	14.8	$0.370	12.6	$0.315	7.3	$0.182
Pump	1.9	0.048	1.8	0.045	1.9	0.048	1.6	0.040	1.0	0.025
Room Unit Fans	15.3	0.381	5.3	0.134	5.3	0.134	5.3	0.134	3.1	0.078
Kitch. Vent. Fan	0	0.0	0	0.0	1.1	0.027	1.1	0.027	0.6	0.015
Total Power	33.5	0.836	20.7	0.519	23.1	0.579	20.6	0.516	12.0	0.300
	Cu ft		Cu ft		Cu ft		Cu ft		Cu ft	
Cooling Water	83.0	0.303	73.3	0.269	79.5	0.292	67.5	0.248	39.1	0.146
Total Operating Cost		$1.139		$0.788		$0.869		$0.764		$0.446
Power Percent of Total		73.2		66.0		66.5		67.5		67.4

In each case the cost of the electrical energy used to operate the cooling system amounted to approximately 65 to 70 percent of the total operating cost, with the cost of cooling water making up the balance. Tests A and B show that reducing the number of room units in operation from 6 to 3 reduced the daily operating cost from $1.139 to $0.788. Tests B and C show that the operation of the kitchen ventilating fan increased the cost of operation of the cooling system by about 10 percent.

The operating conditions for Tests C and D were the same except that the average outdoor temperature between 10:00 p.m. and 7:00 a.m. was 75.6 F in Test C and 69.4 F in Test D. The temperature during the daylight hours was the same for both tests. The cooler night air brought into the house by the ventilating fan during Test D reduced the cooling requirements during the night and early morning sufficiently to effect a net reduction in daily operating cost of about 12 percent as compared to Test C.

A comparison of Tests D and E reveals that opening the windows from 10:00 p.m. to 7:00 a.m. reduced the cost of operation from $0.764 to $0.446 per day.

The foregoing cost figures are only approximate as the shortness of the testing season permitted the running of only one or two tests for each operating condition, and this was not sufficient to average out such effects as day-to-day variations in wind, sun intensity, and outdoor humidity.

CONCLUSIONS

The tests made in the *I=B=R* Research Home during the summer of 1953 have indicated that:

1. The Research Home could be satisfactorily cooled in summer using a water chiller and room units designed to use chilled water in summer and hot water in winter.
2. The same piping system may be used for both the summer and winter phases of operation providing it is covered with a vapor proof insulation to prevent sweating when chilled water is being circulated.
3. With a maximum outdoor temperature of 100 F, the measured maximum sensible cooling load was about 17 percent less than the maximum sensible load determined by conventional methods of estimating cooling loads.
4. Most of the differences between the estimated and observed maximum cooling loads could be attributed to the method of estimating heat gains through glass areas.
5. In estimating cooling loads it is assumed that all radiant energy transmitted through glass is immediately available to heat the air in the room. Actually it first heats objects in the room, and they in turn heat the room air by convection. This results in a time lag and a reduced rate of heat release to the air.
6. As long as room doors are not closed, a satisfactory cooling job could be done in the entire Research Home by operating cooling units in only two or three rooms.
7. With the type of equipment used in these tests, the installation of room units having a cooling capacity in excess of the maximum required decreases the quality of performance from the standpoint of comfort and increases the cost of operation.
8. For satisfactory results the capacity of the water chiller must be sufficient to quickly reduce the temperature of the water in the system to the minimum value.
9. Opening windows at night had no effect on the maximum cooling load, but except for a very few days in Urbana, this practice results in equal or better comfort conditions in the house than obtained by operating with the windows closed at all times.
10. Opening windows at night reduced the cost of operation by as much as 50 percent.
11. Operating a kitchen exhaust fan having a free air delivery of 430 cfm increased the maximum cooling load by about 25 percent and increased the daily cost of operation by about 10 percent.
12. From the standpoint of comfort there appeared to be no need of mechanical ventilation for the test conditions. Some such ventilation might be required during periods when foods are being cooked or at times of unusual occupancy or heavy smoking.
13. The daily operating cost for a design day varies from $1.139 to $0.446 depending upon the method of operation. The highest operating cost was obtained when using all six room units and operating with closed windows at all times. The lowest operating cost was obtained when using only three room units and operating with the windows open from 10:00 p.m. to 7:00 a.m.

DISCUSSION

A. E. DIEHL,* York, Pa. (WRITTEN): We note that this paper is rather definite in its conclusion that actual measured cooling loads were considerably below the calculated loads obtained using THE GUIDE.

We believe that information of this type, along with *A.R.I.* load calculation contained in Standard 6-10 should be used to convince the necessary authorities in the Society that THE GUIDE figures should be abridged or extended inasmuch as they apply to residential cooling loads.

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* Chief Engineer, Commercial Products, York Corp.

Subject paper also contains some rather pertinent information with respect to operating costs obtained in a cooling season and definitely outlines the effect of ventilation, outside air circulation during the night, etc.

Some of the information contained in this paper might be considered in the paper on operating costs,† which was based on the fact that all windows would be closed throughout the entire 24-hr period of the day.

It appears that the authors have given rather conclusive data with respect to the effect of opening bedroom windows at night and shutting down the air conditioning system. He has also shown that some diffusion of air conditioning load can be obtained from one room to the next with a resultant rather low variation in dry bulb temperatures throughout the residence by eliminating units in certain areas of the residence.

We believe that all these factors will eventually enter into this business of residential air conditioning and result in the low first cost and low operating cost that will be necessary if the industry is going to really sell a wide number of units.

E. P. PALMATIER and W. A. GRANT, Syracuse, N. Y. (WRITTEN): This is a very interesting paper because it demonstrates clearly that, regardless of the type of system used for a residence, the basic principles of good application practice cannot be compromised without incurring poor results.

Comparing Tests I and II, it is clear that circulation of too much air over the cooling coils results in unacceptably high humidities, and runs up the operating cost as well. From the data, it appears that 750 cfm should be ample, as compared with approximately 1350 cfm used in Table 1. However, the authors infer that the *number* of units operating is significant in relation to the quality of results, and that the fewer the units, the better the conditions.

This inference is incorrect. If the air quantity through each of the six units had been reduced to the correct value, good conditions would have been possible.

Correct conditioned air quantity to match the load, however, is only part of the story. In any system where cycling of the refrigeration equipment permits excessive rise of the refrigerating medium temperature during the off-cycle, high humidities are inevitable. This is so well known that it is surprising that the authors fail to point it out. The most straightforward way to insure satisfactory conditions of temperature and humidity in a residential cooling system is to match both air cooling capacity and refrigeration capacity to the load so closely that the equipment runs continuously during the hotter weather such as is represented in Fig. 5. Yet the bar chart of compressor operation shows less than 50 percent on-time during daytime and evening, and only about 30 percent on-time over the entire 24-hr period. With such gross oversizing of equipment with respect to the load, humidities of 60 to 75 percent, as actually occurred, can be expected.

We would like to express a word of caution concerning opening windows for night cooling. The tests show average indoor night temperatures 4 to 5 deg *above* outdoors with windows open. This is fine for the tests cited where outdoor night temperatures dropped to 66 or 67 F—but that represents a 28 deg drop from the daytime maximum of 95 F. In most parts of the country, such a drop can certainly not be counted on. With outdoor night temperatures of 75 to 85 F, the indoor temperature will be far from comfortable.

The statement that the ventilation air increased the maximum load by only 25 percent is of considerable interest. If this load is calculated for the design outdoor conditions in Urbana, it amounts to 11,400 Btu/hr latent and 9,700 Btu/hr sensible, or a total of 21,100 Btu/hr. If this were included in the design load calculations, it is evident that nearly 2 tons of refrigeration would be assigned to handle the outdoor

† The Operating Cost of Residential Cooling Equipment, by S. F. Gilman, L. A. Hall and E. P. Palmatier (A.S.H.V.E. TRANSACTIONS, Vol. 60, 1954, p. 525).

air load alone. If the sensible load due to the outside air was increased by only 25 percent, then only about 20 percent of the air exhausted became part of the air conditioning load. This means that something like 90 cfm was pulled in through cracks in the bedrooms, living room, and probably also the dining room. The remaining 350 cfm was evidently drawn in through cracks around the kitchen window as well as around the door leading to the garage, and was discharged before its heat was absorbed by the room air. This implies that for a home of a certain "tightness" there is an upper limit to the amount of outside air load which should be included in the load calculations irrespective of the size of the exhaust fan. It is hoped the authors can be encouraged to pursue this important load component further.

The authors are to be commended for bringing to the attention of our membership important facts about the actual operation of residential cooling equipment.

F. W. LEGLER, Collingswood, N. J.: When you consider the high cost of water, in this case about 25 percent of the total operating cost, has Professor Harris made any study at all as to how that water may be used for sprinkling in a residence, thereby reducing the total cost of water to the home owner?

A. B. NEWTON, Wichita, Kans.: I have two questions I would like to have Professor Harris answer. In the tests run with only a small capacity in the room units, I am wondering just what type control there was on the water chiller.

TABLE A—AIR TEMPERATURES AT CENTER OF LIVING ROOM

LEVEL	AT START OF COOLING CYCLE	AT END OF COOLING CYCLE
3 in. below ceiling	77.0	75.0
30 in. above floor	76.5	74.0
3 in. above floor	76.5	73.5

Air circulation by Living Room unit = 165 cfm.

In other words, you probably had less capacity on the room units than in the water chiller and I am interested in the variations in the water temperature and how it might have affected the tests.

I would also be interested to know how the condensate was measured so that the quantity could be properly correlated with the hours of operation for any given test.

I. S. DANE, Boston, Mass.: I would like to ask Professor Harris about the distribution of air within the room; *i.e.*, what was the velocity of the air leaving the units and what was the stratification of temperature in relation to the velocity.

AUTHORS' CLOSURE (W. S. Harris): No attempt was made to accurately measure air movement at different locations within the rooms during the tests. Occupants of the house noticed no objectionable air motion except in close proximity to the discharge grille of the room unit. The fans in the room units were operated continuously and each unit circulated approximately 170 cfm. Typical air temperature variations in the living room are given in Table A. These variations were representative of the other rooms as well and indicate sufficient air movement to prevent undesirable temperature stratification.

As the number of room units in operation was reduced there was a corresponding reduction in the total load on the water chiller. When only two room units were in operation the compressor motor cycled by action of the low side limit control. This control stopped the compressor when the temperature of the water leaving the chiller

dropped to 40 F and re-started the compressor when this water temperature rose to 45 F.

The water removed from the air by each room unit was collected in separate containers. The amount collected was recorded periodically during each test. For each period it was assumed that the rate of removal of water from the air was constant and equal to the total weight of water removed by the room unit during the period divided by the total operating time of the water circulating pump during that same period. Operation of the water circulating pump was controlled by the room thermostat.

Investigating the possible uses of cooling water discharged from the condenser was outside the scope of this study. It is possible that it could be used for irrigation or other such uses, but it is the opinion of the authors that it is more important to reduce the total water consumption by employing the use of cooling towers or similar devices or by developing air cooled equipment suitable for residential use.



1519

No. 1519

ANALOGUE COMPUTER ANALYSIS OF RESIDENTIAL COOLING LOADS

By T. N. Willcox,* C. T. Oergel,** S. G. Reque,†
C. M. toeLaer,†† and W. R. Brisken,△ Bloomfield, N. J.

PERSISTENT reports from various parts of the country indicated that many homes are being comfortably cooled with smaller equipment than would be required to match the calculated sensible heat gain. Three factors could contribute to this condition:

1. A fundamental error may exist in the heat gain calculation for residences.
2. Users may be satisfied with higher indoor temperatures than the design temperature assumed for the load calculations.
3. Equipment output may be higher than the manufacturer's rating.

The likelihood that cooling equipment output could run consistently higher than published performance ratings is remote because of the substantiation of published ratings by accurate laboratory tests. Consideration was given, however to the low indoor wet-bulb temperatures which in effect increase the sensible capacity of the equipment. This investigation was limited to performance preferences of users and to residence cooling loads.

User Preference

During the summer of 1953 extensive comfort surveys in 17 homes were completed to substantiate a six-year field investigation. Pertinent data concerning these homes are given in Table I. Each survey included a personal interview, temperature measurements in the home, a complete construction detail for a room by room heat gain calculation, numerous trips to the various homes, and an end of season follow-up questionnaire.

Selection of the test homes was made on the basis of climate, the type of house, and the amount of undersizing of cooling equipment based on existing methods of load calculation. Single family, owner-occupied houses were selected. Floor area ranged from 1400 to 3600 sq ft. Brick veneer, frame, and masonry houses were included. Some had wall insulation and all had ceiling insulation.

Dallas, Texas, and Kansas City, Mo., were chosen as areas having a high outdoor design temperature and a medium daily temperature range. Northern

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Swampscott, Mass., June 1954.

New Jersey (15 miles from New York City) was chosen for a low daily temperature range. Design temperatures in common use are shown in the following tabulation:

	Outdoor Dry-Bulb Temp[1] F	Outdoor Wet-Bulb Temp[1] F	Daily Range Temp[2] F Deg
Dallas	100	78	Medium
Kansas City	100	76	Medium
Newark	95	75	Low

[1] Exponent numerals refer to References.

Table 2 gives the results of heat gain calculations on the individual houses, the indoor temperature range which the user desired, and the degree of satisfaction obtained. The most interesting thing regarding these results is the user acceptance of a surprising degree of *undersizing* judged by current standards of heat gain. The lowest satisfactory percentage of installed capacity for a whole house was 62 percent.

Although the temperature ranges of user comfort appear to be widely scattered, the results assume significance when the high and low limits are plotted on probability paper. Fig. 1 shows that 50 percent of these and similar users

TABLE 1—TEST HOUSE INFORMATION

House No.	Location	No. of Stories	Type of Foundation	Wall Construction[a]	Total Sq Ft Floor Area Conditioned	No. of Occupants	Children 15 Years or Under
1	Dallas	1	Crawl	B.V.I.	1705	2	No
2	Dallas	2	Crawl	Frame	3630	5	Yes
3	Dallas	1	Crawl	B.V.I.	1485	5	Yes
4	Dallas	1	Crawl	B.V.I.	1850	3	No
5	Dallas	1	Crawl	B.V.	1655	4	Yes
6	Kansas City, Mo.	2	Basement	Frame, I	2105	4	Yes
7	Kansas City, Mo.	2	Basement	C.B.I.	2290	4	Yes
8	Kansas City, Mo.	2	Basement	Frame	1690	2	No
9	Kansas City, Mo.	1	Basement	B.V.	1410	2	No
10	Kansas City, Mo.	2	Basement	Frame	1480	4	No
11	New Jersey	1	Basement	Frame, I	1745	4	Yes
12	New Jersey	1-SL[b]	Basement	Frame, I	1945	6	Yes
13	New Jersey	2	Basement	Frame	1780	4	Yes
14	New Jersey	1	Basement	B.V.	1616	4	Yes
15	New Jersey	1-SL	Basement	Frame, I	1500	2	No
16	New Jersey	1-SL	Basement	Frame, I	1930	4	Yes
17	New Jersey	2	Basement	Frame, I	1545	6	Yes

[a] B.V.I. = Brick Veneer, Insulated
C.B.I. = Cinder Block, Insulated
[b] S.L. = Split Level

would accept 81 F as a maximum and 75 F as a minimum comfortable indoor temperature. Also, Fig. 1 shows that for an indoor temperature of 78 F, approximately 90 percent of these and similar users would be satisfied. The rest would be equally divided, 5 percent too hot and 5 percent too cold.

The field test survey makes it clear that the interviewed users prefer an indoor temperature in the range customarily used for heat gain calculations. This same indoor temperature, when substituted into the heat gain calculation, results in the selection of cooling equipment substantially larger than it need be for comfort to these users. Serious doubt is thus cast upon the accuracy of current methods used in calculating residential heat gains.

Obviously the inaccuracy is due to failure to consider the heat storage capacity of the structure and contents.

The existence of these thermal storage effects has been recognized for some time. Various methods based on experience or theory or a combination of the

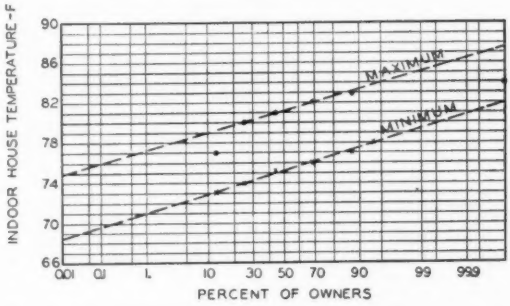


FIG. 1. MAXIMUM AND MINIMUM INDOOR TEMPERATURE LIMITS ACCEPTABLE TO OWNERS

two have been devised for taking them into account. Some ingenious methods make use of time averages of heat gains. These methods all leave something to be desired, and a more realistic solution to the problem would be appreciated. The analogy between heat and electric circuits has threfore been used as a means of solution.

First attempts were to instrument a house and devise a circuit to fit the test results. Mathematical difficulties encountered were of two kinds. If the assumed house circuit was simple enough to be fitted to the test results, its parameters could not be related to house details. The circuit could not then be used to predict performance in a house of different construction. If the circuit was sufficiently complex to represent the house, it could not be fitted to the test data. There were too many constants to be evaluated. Many complex circuits can be constructed to produce the same response. Another difficulty in fitting the circuit to the test results is that, with varying load conditions, there are always twice as many constants as equations to determine them. A second set of equations could be obtained from operation at constant conditions. Operation under constant conditions would determine the thermal resistances and operation under varying conditions would determine the thermal storages. Constant operating

conditions are, however, very rare in nature if they exist at all. It means that the weather conditions must be constant and that the equipment capacity must be precisely suited to the load. This attempt at the problem solution was abandoned.

With the advent of the electrical analogue computer, the problem was attacked again. This time the process was reversed. A circuit was devised based on the physical structure of a house. Mathematical difficulties were overcome by using an analogue computer. The circuit was established for a house which had been tested.[3] Computer results agreed with test results within engineering accuracy. This was accepted as verification of the circuit.

THERMAL CIRCUIT

The thermal circuit developed for these calculations is shown in Fig. 2. It comprises five major branches and a thermostat branch, each of which is connected to the bus, the voltage of which corresponds to indoor temperature, which was assumed to be the same throughout the house. One branch shown at the upper left of Fig. 2 consists of the straight resistance loads. Actually the windows, and doors and Venetian blinds have some thermal capacity. They are here neglected because their time constants are small being of the order of 3 min, while the time constants of the other branches vary from 2.4 hr for frame

TABLE 2—TEST HOUSE RESULTS

HOUSE NO.	NOMINAL INSTALLED TONS	INSTALLED CAPACITY PERCENT OF LOAD[a] CALCULATED	USER COMFORT RANGE[b]		USER SATISFIED	COMMENT
			High F	Low F		
1	3	60	80	76	Yes	Partial cooling demand
2	7½	103	—	—	Yes	
3	3	79	83	75	Yes	
4	5	75	77	73	Yes	
5	3	44	84	75	Yes	Partial cooling demand
6	5	134	84	77	No	Blocked Supply One room
7	3	62	82	76	Yes	Masonry Partitions and floors
8	3	75	81	78	Yes	
9	3	83	83	76	Yes	
10	3	78	83	81	No	Temp. good, Too noisy at night
11	2½	74	—	—	No	Hot room over garage
12	3	75	77	74	No	Hot room over garage
13	2	56	82	77	No	Upstairs too hot
14	2	79	82	74	Yes	
15	2	68	82	77	Yes	Marginal
16	3	64	81	76	No	Hot room over garage
17	3	86	80	75	Yes	

[a] Calculated loads are based on 80 F indoor design temperature and include duct allowances.
[b] Temperatures corrected for error in house thermometer on thermostat.
Users were asked:
How high (and low) can the indoor temperature go and still feel reasonably comfortable?

walls to 9 hr for solid masonry walls. The time constant is taken as the time required for the heat flow to the house air to reach 63.2 percent of its steady state value when the branch is subjected to an instantaneously applied constant outdoor temperature. The infiltration can be taken as a resistance because only sensible heat gains are being considered. For the case of Venetian blinds, the sun load coming through the glass is introduced on the blind. The resistance R_2 then consists of three separate resistances, outside air to glass, glass to air between the window and the blind, and air to blind. The resistance R_3 is from the blind to the indoor air. The resistances through the glass and the Venetian blinds are neglected since they are small compared to the air film resistances. When there are no Venetian blinds, the sun load through the glass is introduced on the floor, then R_2 consists of two film resistances, outside air to glass and glass to inside air, and R_3 is of course zero. The resistance R_4 consists of outside and inside air film resistances.

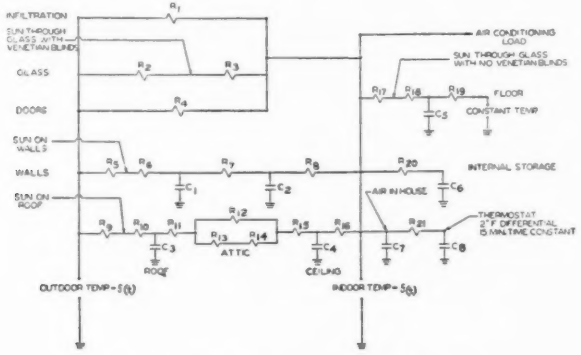


FIG. 2. EQUIVALENT THERMAL CIRCUIT

The branch for the walls is a single π branch with the sun load introduced at the outside wall surface. The use of a single composite wall to replace the four separate walls is justified on a theoretical basis. It is to be noted that the differential equations giving the wall heat flows to the indoor air on a unit area basis, are identical for the four walls except for the solar radiation heat loads. These four equations can be combined to give the composite wall which will give, when subjected to the total solar radiation, for the four separate walls, the correct total heat flow into the house. The equality which holds for heat flow does not hold for the temperatures in the wall. The composite wall is therefore used here only for heat gain calculations. R_5 is the outside air film resistance. R_6 is one half the resistance of the outside wall section, that is that part of the wall which is outside the studding. C_1 is the thermal capacity of the outside wall. R_7 consists of half the resistance of the outside wall section, the resistance of the air space, and one half the resistance of the inside wall section. C_2 is the thermal capacity of the inside wall section. R_8 consists of one half the resistance of the inside wall section and the air film resistance from wall to inside air. The splitting of the wall resistances puts the thermal storage at the average temperature.

The branch for the roof, attic, and ceiling is actually a π circuit with the attic shown in detail. R_9 is the outside air film resistance. The solar radiation load is introduced at the roof surface. R_{10} is one half the roof resistance. C_3 is the roof thermal storage. R_{11} is the second half of the roof resistance. R_{13} is the film resistance from roof to the attic air. R_{14} is the film resistance from attic air to ceiling. The thermal storage of the attic air should be introduced between R_{13} and R_{14}. It is, however, small so that its time constant is only a few minutes. It is therefore neglected. An infiltration heat load should be introduced between R_{13} and R_{14}. It is negligibly small for natural ventilation of one change per hour. R_{12} is to account for radiation from the hot roof to the top of the ceiling. R_{15} is one half of the ceiling resistance. C_4 is the thermal storage of the ceiling. R_{16} is the other half of the ceiling resistance and the film resistance from ceiling to inside air.

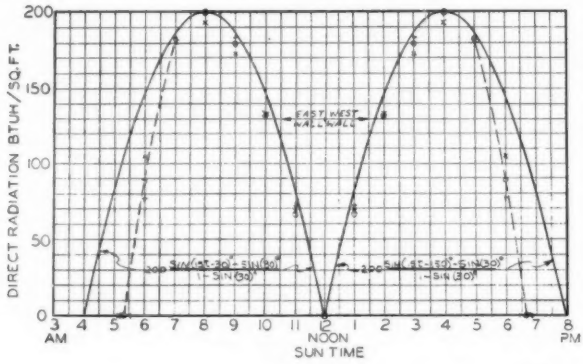


FIG. 3. SOLAR RADIATION INTENSITY FOR EAST AND WEST WALL

At the upper right of Fig. 2 a T circuit represents the floor. R_{17} is the film resistance from inside air to floor surface. For the cases of no Venetian blinds the sun load coming in through the glass was introduced at the floor surface. R_{18} is one half the floor resistance. C_5 is the thermal storage of the floor. R_{19} is the other half of the floor resistance and the film resistance from floor to basement air.

The fifth branch is the internal storage L circuit. It comprises internal partitions, furniture and all thermal storages which have heat exchange only with the air in the house. C_6 is the total thermal storage and R_{20} is the resistance from air to average storage temperature and hence includes the air film resistance.

At the bottom right of Fig. 2 is shown C_7 which is the thermal storage of the air in the house and an L circuit comprising R_{21} and C_8 to represent the thermostat with a 15-min time constant. The thermostat was arranged to control the air conditioning load, shown at upper right, with a 2 deg differential.

The parameters for this analogue circuit determined from construction details, and they are therefore related to physical entities.

The circuit parameters were determined for the Research Residence No. 2 of

the University of Illinois. The required cooling capacity as determined by analogue circuit calculations was only 7 percent higher than that shown by an actual test on the house.[3] The imposed conditions of air and sun for these calculations corresponded to those prevailing during the test. This check was accepted as additional evidence that the analogue circuit represents the house with good engineering accuracy.

THERMAL LOADS

Three sources of heat were considered; that from the outdoor air, that from solar radiation, and that due to temperature difference between house and basement air temperatures. No consideration was given to heat from occupants or

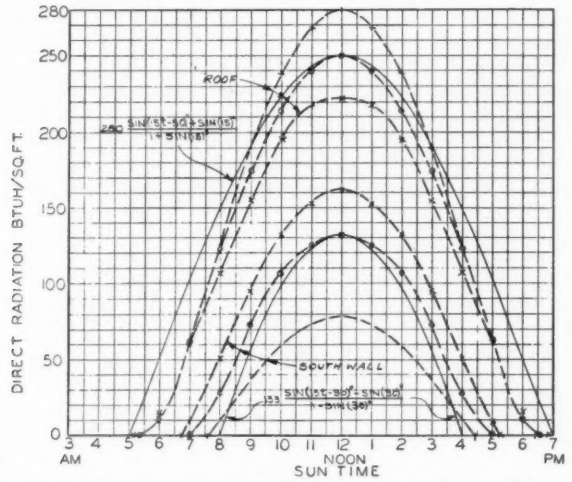


FIG. 4. SOLAR RADIATION INTENSITY FOR SOUTH WALL AND ROOF

such indoor loads as appliances, lights, etc. Only the sensible components of the heat gains were considered. Dehumidification loads were circumvented by assuming each ton of refrigeration to consist of 8600 Btuh sensible and 3400 Btuh latent heat. This average is in accord with experience.

Outdoor air temperature was taken as a sine curve with the maximum at 2:00 p.m. sun time, and it was assumed to repeat itself day after day. Three sine curves were used as representing different climates. One varied from 85 F to 100 F which could represent Tampa, Fla., with 28 deg latitude. One varied from 75 F to 100 F which could represent Cincinnati, Ohio, with 40 deg latitude. One varied from 55 F to 100 F which could represent Spokane, Wash., with 48 deg latitude.

The sun loads which were taken for a clear sky on August 21 consisted of direct and diffuse radiation on walls and windows and direct radiation on the roof. Fig. 3 shows direct radiation intensity on the east and west walls and Fig. 4 shows direct radiation intensity on the roof and south wall. Dashed

curves are drawn through the calculated intensities. The solid curves are portions of sine waves and show the intensities actually used in the calculations. Latitude has a negligible effect on east and west wall intensities. Latitude does affect roof and south wall intensities but these two loads have a small effect on the house. The south wall load is small and the large roof load fails to penetrate the attic and the ceiling insulation. If the solid curves were not considered sufficiently accurate it would therefore be relatively easy to correct for latitude. Diffuse radiation intensity was taken proportional to roof radiation intensity with a peak intensity of 25 Btuh per sq ft. The sun loads are the product of the intensity, the incident area, and a coefficient. The radiation transmission coefficient for glass was taken at 0.9. The absorption coefficients were 0.5 for (white) frame walls, 0.75 for brick veneer and heavy masonry, and 0.9 for the roof.

The basement was taken as an infinite heat reservoir at 80 F. For awnings it was assumed that the solar load through the windows was just half of that which was calculated for no protection. The analogue circuit was modified for the case of Venetian blinds which were assumed drawn and closed on all windows.

The thermostat was arranged to have a 15-min time constant and a 2 deg differential. Three thermostat settings were used: 70 to 72, 74 to 76, and 78 to 80 F.

Air infiltration was taken at one air change per hour in all cases.

COOLING LOADS BY ANALOGUE COMPUTER METHODS

It has been recognized for a long time that a complete duality exists between resistance-capacity electrical networks and thermal circuits. If it is assumed that voltage corresponds to temperature and electrical charge corresponds to quantity of heat, a consistent equivalence results. Current, which is rate of change of charge, then corresponds to rate of flow of heat, electrical resistance corresponds to thermal resistance, and electrical capacitance corresponds to thermal capacity. It is impracticable to construct equivalent electrical networks for thermal circuits which show very slow behavior with time. However, it is easy to construct electrical networks which differ only in the ratio of their time responses. The technique of solution using this approach is very direct. An electrical network is constructed on a suitable time scale. Electrical signals of the same form as the independent heat sources are applied to the proper terminals of the network and the response of the network is recorded, by oscillograph, at the points under consideration. The practical limitations of this method of study appear to lie in the techniques of signal generation and recording. This method has many practical advantages. It was not used in these studies, however, because an analogue computer was available with which to obtain the desired solutions within the degree of accuracy desired.

For the purpose of this study, an analogue computer may be considered as an assembly of amplifiers which have two significant properties. The first property of interest is that the amplifiers can be made to respond dynamically in a fashion analogous to a circuit element or simple combination of circuit elements. The other property of interest is that the amplifiers can be excited from any number of separate sources. The technique used to set up the problem follows from these two properties. See Appendix A.

If any junction of the thermal circuit is considered in detail, the differential equation of heat balance at that junction will involve a simple time varying

factor and the temperatures of adjacent junctions. For example, the temperature at junction 2, T_2, might be calculated from the heat balance equation as a function of T_1 and T_3 (the temperatures at junctions 1 and 3) and t_2, a time constant. An amplifier in the computer can then be set up with the proper time behavior and gains for the corresponding temperatures.

If the foregoing process is carried out for each of the network junctions, the remaining steps in setting up the thermal circuit are simple. The output of amplifier 2, calculating T_2, is connected to all other amplifiers requiring T_2 as inputs, etc. The resulting interconnection of amplifiers automatically satisfies all of the conditions existing in the original thermal circuit. The solutions for the transient temperatures are now obtained by applying suitable signals to the independent inputs of the thermal circuit. At this point the procedure of signal generation and recording are similar to those which apply to the previously described method of using a static electrical network.

Several advantages are obtained by using the analogue computer to simulate the thermal circuit. The outputs of the computing amplifiers used to solve for the junction temperatures are negligibly affected by external loading. As a result, the individual temperatures may be read with ordinary meters or oscillographic equipment without disturbing the temperature distribution in the circuit. In addition, the inputs to these amplifiers are usually of very high impedance so that the signal generation equipment need not have significant power capability.

There are two principal disadvantages in the use of an analogue computer, one real and one only apparent. The cost of using analogue equipment for simulating thermal circuits may appear to be extremely high. In terms of total cost it is undoubtedly the most economical, because set-up time and running time are held at a minimum. Relatively few comprehensive studies are required to justify an investment of this type. The real disadvantage in using an analogue computer technique of the type described lies in the manner in which errors can accumulate unless extreme care is used in setting gain coefficients. It was found necessary to set individual amplifiers within a few tenths of 1 percent in order to obtain accuracies in overall results of the order of 3 percent. For engineering purposes this order of accuracy is sufficient and can be obtained with most of the analogue computers which are currently available commercially.

A typical set of oscillograms are shown in Figs. 5 and 6. These oscillograms were recorded on standard Brush recording equipment. In this particular study the time scale was arranged so that one second of machine time corresponded to one hour of problem time. One minute of recording therefore gave a 60 hr history on the house.

CALCULATIONS AND RESULTS

There were 108 possible different houses having characteristics shown in Appendix B, which could be subjected to nine possible different sets of conditions giving a total of 972 possible different sets of calculations. A sample of about 5 percent of the total possible combinations was chosen to study various effects. The analogue circuits were set up in the computer and subjected to the daily air temperature and solar load cycles. In each case a record was made of the indoor temperature for installed capacities of ½, 1, 2, 3, and 4 tons. Fig. 5 is representative of these results, showing indoor temperatures for various capacities of cooling unit. In each case the daily weather cycle was repeated a

sufficient number of times so that initial transients were removed. For some of the cases set up for calculation, the daily cyclical heat flows in the five separate branches of the analogue circuit, were recorded along with the on and off operation of the cooling unit. Fig. 6 is representative of these results.

The computer calculated cyclical results. There were analyzed for (1) effect of installed cooling capacity on indoor air temperature, (2) unit heat gains, and (3) an allowance for internal heat storage.

From the accumulated data it is practicable to establish factors which can be used to estimate the separate heat gains for the various surfaces and for the heat storage allowance. The net heat gain then gives the required capacity. Analysis of the machine calculated results shows that the total net heat gain is a maximum at very nearly 3:00 p.m. sun time for all the cases considered. The

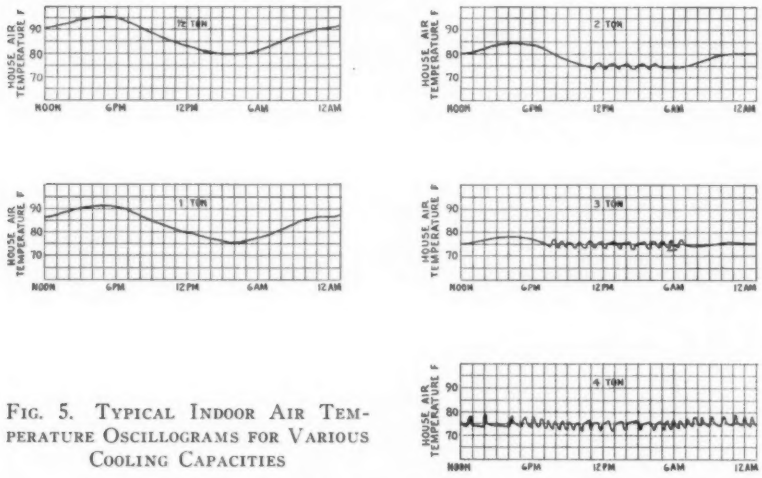


FIG. 5. TYPICAL INDOOR AIR TEMPERATURE OSCILLOGRAMS FOR VARIOUS COOLING CAPACITIES

separate heat gains were therefore determined at 3:00 p.m. These heat gains were used to evaluate the heat gain per unit area shown in Table 3.

The unit heat gains obtained from the analogue study differ from those being used for steady state calculations. The differences, although small, warrant some changes in steady state unit heat gains if accuracy is to be obtained (discussion of these changes is outside the scope of this paper). The big discrepancy between the required capacity and that computed by the steady state method is due to heat storage.

HEAT STORAGE ALLOWANCE FOR COOLING LOAD CALCULATION

The steady state method of calculating heat gains is an attempt to use a static method to analyze a dynamic phenomena. Perhaps the most serious omission is the internal storage which in effect is a heat sink during the heat of the day. The effectiveness of internal storage as a heat sink depends on the time rate of increase of its temperature. This in turn is related to the time rate of increase of indoor air temperature. These relations were investigated. The basic rela-

tionship is that the difference between the house heat gains and the cooling capacity is equal to the sum of the heat being absorbed by the air in the house and the heat being absorbed by internal storage. In symbols:

$$Q_i - Q_o = C_a\,(dT_a/dt) + FC_I(dT_I/dt)$$

where

Q_i = total heat gains
Q_o = air conditioning capacity
C_a = thermal capacity of air in house
T_a = temperature of air in house
F = a factor to be determined
C_I = thermal capacity of internal storage
T_I = temperature of internal storage
t = time

Now it is necessary to relate the time rate of change of internal storage temperature to that of the air temperature. This can be done theoretically by

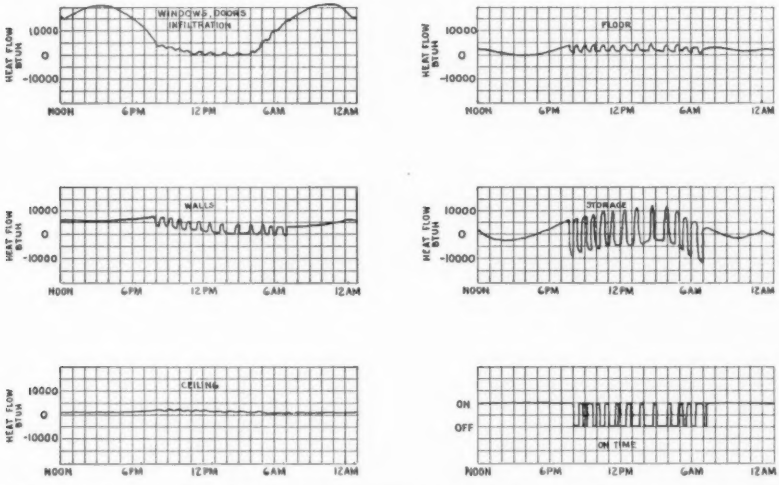


FIG. 6. TYPICAL HEAT FLOW OSCILLOGRAMS FOR THERMAL CIRCUIT BRANCHES WITH 3 TONS OF COOLING

applying a sinusoidal temperature to the internal storage circuit. Neglecting the transient solution, it is found that

$$dT_I/dt = \beta\,(dT_a/dt)$$

where

$T_a = A \cos \omega t$
A = maximum indoor temperature
$\beta = \sin \theta \cos (\theta + \omega t)/\sin \omega t$
$\theta = \tan^{-1} [R/(\omega C_I)]$
R = thermal conductance between internal storage and house air
ω = 360 angular degrees/24 hours when time, t, is in hours

Using this result the heat flow to internal storage can be written

$$FC_I \beta \, (dT_a/dt)$$

The value of this heat flow is known from the machine calculations. The value of dT_a/dt can be obtained from the machine calculated indoor temperature and the value of C_I is known. Thus the product $F\beta$ can be calculated and with engineering accuracy it can be taken as unity. Then

$$Q_o = Q_i - (C_I + C_a) \, dT_a/dt$$

and this equation can be used to calculate the required cooling capacity Q_o for any acceptable rate of increase of indoor air temperature. Q_i is of course deter-

TABLE 3—UNIT HEAT GAIN FOR 22 F DEG TEMPERATURE DIFFERENCE BTUH PER SQ FT

CASE	WALL	CEILING	FLOOR	WEST WINDOW SOLAR	WINDOWS		CONDUCTION AND DIFFUSE RADIATION
					Conduction	Diffuse Radiation	
6B	5.48	1.46	0.73	93.0	11.14	16.96	28.1
7B	6.48	1.35	0.34	96.5	11.14	16.96	28.1
8B	6.17	1.39	0.62	101.3	11.14	16.96	28.1
8M	6.33	1.60	0	94.2	11.14	16.96	28.1
8O	6.03	0.79	0.83	86.0	11.14	16.96	28.1
8P	6.07	1.57	0.29	106.6	11.14	16.96	28.1
8R	6.01	1.34	0.48	107.7	11.14	16.96	28.1
9B	6.92	1.82	0.78	101.4	11.93	17.27	29.2
9N	6.26	—	0	98.3	11.93	17.27	29.2
By Computer, Avg.	6.19	1.42	0.45	98.3	11.32	17.03	28.4
Steady State	6.9	4.0	0	75.	—	—	28.

Note: All heat gains based on fully shaded windows.

mined from the heat gain factors. It is noted that C_a is usually small compared to C_I and may then generally be neglected.

APPLICATION OF HEAT STORAGE ALLOWANCE

Heat storage capacity of interior partitions, floors, and furnishings can be used to reduce the load on air conditioning equipment. Two conditions are necessary:

1. The surface temperature of the heat storage body must be cooler than the room air.
2. Warm room air must be moved to the cooling surfaces.

The first condition requires the indoor temperature to rise above the preset thermostat temperature. If the rise is kept small enough, no perceptible change in comfort will be noticed. The maximum temperature rise to be permitted by the *Heat Storage Allowance* herein described is 2 F deg.

The second condition requires rapid redistribution of heat from any localized sources to all the rooms of the house. Instantaneous loads from sunlight, people, and appliances may quickly saturate the allowable heat storage capacity within the rooms where they occur, after which the air temperature in those rooms

TABLE 4—HEAT STORAGE ALLOWANCES

ROOM BY ROOM CALCULATION		OVERALL CALCULATION		
No. of Heat Storage Surfaces	Percent Reduction of Conduction Plus Infiltration	House Foundation	No. of Stories	Percent Reduction of Conduction Plus Infiltration
		Crawl	1	5
1	0	Crawl	2	10
2	5	Slab	1	10
3	10	Slab	2	15
4	15	Basement	1	10
5	20	Basement	2	15

Note: Heat storage surfaces have conditioned space on both sides. Slab floors are heat storage surfaces.

becomes uncomfortable. Because of the variety and specialized nature of instantaneous loads, they are best handled by providing a full match of cooling capacity in the air conditioner. Other loads, such as conduction and infiltration are distributed all over the house, just as is the heat storage capacity. These are the loads which can be most easily reduced by heat storage.

The analogue computer study gave the following ratios:

$$\text{For 1 story house } \frac{\text{Heat Storage Rate}}{\text{Conduction \& Infiltration Load}} = 0.13 \text{ or } 13\%$$

$$\text{For 2 story house } \frac{\text{Heat Storage Rate}}{\text{Conduction \& Infiltration Load}} = 0.22 \text{ or } 22\%$$

These ratios define the term *Heat Storage Allowance.*

Practical use of the heat storage allowance is made by subtracting from the conduction plus infiltration load a percentage equal to the heat storage allowance. The allowance is related to the number of heat storage surfaces in a room, there being a maximum of six, counting the four walls, ceiling and floor. A heat storage surface has conditioned air on both sides. Concrete slab floors

TABLE 5—RELATIONSHIP OF HEAT STORAGE TO HEAT GAIN BY CONDUCTION AND INFILTRATION

TEST HOUSE NO.	NO. OF STORIES	FOUNDATION TYPE	CONDUCTION PLUS INFILTRATION	SPECIFIC HEAT METHOD		OVERALL METHOD		ROOM BY ROOM METHOD	
				Heat Storage for 1 deg F Rise	Storage in Percent of Cond and Infil	Heat Storage Rate	Storage in Percent of Cond and Infil	Heat Storage Rate	Storage in Percent of Cond and Infil
			Btuh	Btu	Percent	Btuh	Percent	Btuh	Percent
2	2	Crawl	51,600	9,200	17.8	5,160	10	4,630	9.0
3	1	Crawl	26,920	2,560	9.5	1,350	5	1,620	6.0
8	2	Basement	30,670	8,590	28.0	4,600	15	3,270	10.7
9	1	Basement	24,760	7,800	31.5	2,480	10	2,930	11.8
11	1	Basement	25,820	8,100	31.4	2,580	10	2,000	7.7
13	2	Basement	22,100	10,200	46.2	3,315	15	2,565	11.6

are, also, considered to be heat storage surfaces, since they have such a large thermal storage mass away from the exposed perimeter.

Table 4 gives the percentage heat storage allowances by which conduction and infiltration loads are reduced when making a room-by-room heat gain calculation on a residence. The same tabulation gives heat storage allowances for an overall load calculation, classifying residences by number of stories and type of foundation. A comparison of the heat storage rates calculated by these two methods is included in a portion of Table 5. This table also shows for comparison the amount of heat that could be stored in the interior partitions and floors of the same houses if their temperatures were permitted to increase 1 F deg.

The recommended heat storage allowances should be used with caution since the conduction unit heat gains used to calculate conduction load must be consistent with those found by the analogue computer analysis.

TABLE 6—TEST HOUSE LOADS CALCULATED WITH HEAT STORAGE ALLOWANCE

HOUSE NO.	STEADY STATE INSTALLED CAPACITY	HEAT STORAGE METHOD INSTALLED CAPACITY	CAPACITY SATISFACTORY
	Percent	Percent	
1	60	74	No
2	103	112	Yes
3	79	84	Yes
4	75	82	Yes
5	44	—	—
6	134	—	—
7	62	77	Yes
8	75	88	Yes
9	83	99	Yes
10	78	84	Yes
11	74	80	No
12	75	81	No
13	56	61	No
14	79	87	Yes
15	68	83	No
16	64	78	No
17	86	108	Yes

Note: Heat gains calculated room by room. Indoor temperature 80 F. Duct allowances included.

Attention is called to the conservative size of the heat storage allowances in Table 4 as compared to the size of the allowances found in the analogue computer study. The allowance for a one story crawl space house is given as 5 percent which is less than half of the 13 percent determined for a one story house with heat gain through the floor. The allowance for a two story crawl space house is given as 10 percent which again is less than half of the 22 percent which was determined by the computer study.

The total sensible heat gains of the field test houses were recalculated using the room by room heat storage allowances. Table 6 gives the results as a percentage of nominal installed capacity. In all houses with 84 percent or higher installed capacity the owners were satisfied. However, the differences in occupants' desires indicate an ample reserve in the equipment selection. By contrast similar calculations omitting the heat storage allowance show that all houses with 76 percent or higher capacity had sufficient cooling.

If the heat storage allowances given in Table 4 had been increased to the full extent indicated by the analogue computer, the ratio of installed cooling capacity to calculated heat gain would have been over 90 percent for satisfactory cooling capacity. It is not recommended that full advantage of heat storage effects be taken at this time. Some margin for error is still needed to offset imperfectly balanced air distribution systems. The heat storage allowances recommended make adequate provision for normal internal loads.

The inherent characteristics of family usage of a home along with the external elements which result in a slight unbalance in temperatures between rooms, and possible variations in construction, indicate a definite need for a reserve allowance. As experience is gained in sizing equipment by the heat storage method, further refinement can be made in the heat storage allowances.

CONCLUSIONS

1. An electrical analogue computer can effectively solve the transient heat flow equations for a house.
2. Error in conduction, solar and infiltration heat gain factors does not account for the successful cooling of some residences with undersized equipment.
3. Heat storage capacity of unexposed interior partitions, floors, and ceilings is the reason for successful cooling with smaller sized equipment than needed to match the load calculated by the steady state method.
4. A reliable and practical relation has been established between house construction and heat storage.

REFERENCES

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2. Cooling Load Calculation, Manual No. 11, 1953, p. 31 (*National Warm Air Heating & Air Conditioning Association*).
3. Cooling a Small Residence with a Two-Horsepower Mechanical Condensing Unit, by H. T. Gilkey, D. R. Bahnfleth and R. W. Roose (A.S.H.V.E. TRANSACTIONS, Vol. 59, 1953, p. 283).

ACKNOWLEDGMENT

The authors gratefully acknowledge the assistance of Alvin Nagler, Air Conditioning Division, General Electric Co., Bloomfield, N. J., who calculated the coefficients for the thermal circuits.

APPENDIX A

Description of Analogue Computer Technique

The analogue computer technique used in these studies was based on the simple heat balance equation. On this basis the conversion of the thermal circuit to an analogue computer diagram follows a direct course which will be outlined.

Consider first a typical junction in the thermal circuit and its adjacent junctions, Fig. A-1. Junction (2) is the junction under consideration. Junction (1) and (3) are the adjacent junctions and may have thermal resistances and thermal capacities associated with them but not shown. R_{12} represents the thermal resistance between junctions (1) and (2) while R_{23} represents the thermal resistance between junctions (2) and (3). C_2 represents the thermal capacity of junction (2).

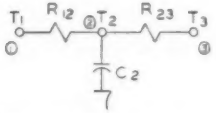


FIG. A-1. TYPICAL JUNCTIONS IN A THERMAL CIRCUIT

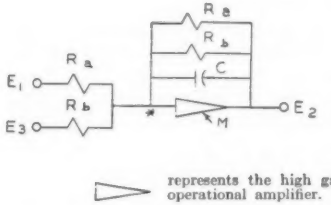


FIG. A-2. ANALOGUE COMPUTER CIRCUIT

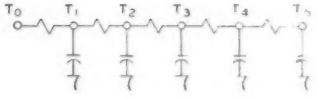


FIG. A-3 *(left)*. THERMAL CIRCUIT

The equation for heat balance at junction (2) may be written as follows:

$$\frac{T_1 - T_2}{R_{12}} + \frac{T_3 - T_2}{R_{23}} = C_2 \frac{dT_2}{dt}$$

Using the operator notation, $p = d/dt$, and transposing the equation, this becomes

$$T_2\left(\frac{1}{R_{12}} + \frac{1}{R_{23}} + C_2 p\right) = \frac{T_1}{R_{12}} + \frac{T_3}{R_{23}}$$

This equation is now in such a form that it may be directly identified with a simple analogue computer circuit.

A suitable analogue computer circuit is shown in Fig. A-2.

The junction marked * in Fig. A-2 is frequently called the summing point. The operational amplifier is a high gain d.c. amplifier, sign reversing, with negligible input current drain. If the gain of the operational amplifier is sufficiently high the voltage which exists at the summing point, *, is always completely negligible with respect to E_1, E_2, and E_3. The current balance equation for the summing point may be written as follows:

$$\frac{E_1}{R_a} + \frac{E_3}{R_b} = -E_2\left(\frac{1}{R_a} + \frac{1}{R_b} + Cp\right)$$

With the exception of the change in sign, this equation has the same form as the heat balance equation derived for the thermal circuit. The resistances of the computer circuit may be directly identified with the thermal resistances. The capacitor of the computer circuit may be identified with the thermal capacity. The voltages of the analogue circuit may be identified with junction temperatures.

Thermal resistances are customarily written in terms of reciprocal Btuh. The electrical units correspond to a relation where resistance is written in terms of reciprocal coulombs/sec. A simple direct conversion from thermal to electrical units will result in one hour of thermal problem time corresponding to one second of analogue computer time. Since this time scale was convenient it was used in the studies described. The conversion factors used are as follows:

	Thermal	*Electrical Analogue*
Resistance	1 unit = 1 F deg/1000 Btuh	1 unit = 1 megohm
Capacity	1 unit = 1000 Btu/1 F deg	1 unit = 1 microfarad
Time	1 unit = 1 hour	1 unit = 1 second
Temperature	1 unit = 1 F deg	1 unit = 1 volt

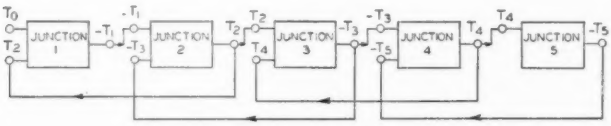


FIG. A-4. COMPUTER DIAGRAM

The total analogue computer circuit for a thermal structure consists only of the interconnected operational units for the individual junctions. Fig. A-3 shows a typical circuit involving a temperature source, T_o, and a five junction structure (assumed insulated at the far end). Fig. A-3 shows the interconnection of operational units for the structure. The operational units would have diagrams as shown in Fig. A-4.

APPENDIX B

House Characteristics

Wall Constructions were:

1. Frame consisting of 10-in. bevel cedar siding, 35 lb building paper, 25/32-in. shiplap sheathing, 2 × 4-in. yellow pine studs on 16-in. centers, 3/8-in. decorated gypsum board.
2. Brick veneer consisting of 4-in. brick, 35 lb building paper, 25/32-in. shiplap sheathing, 2 × 4-in. yellow pine studs on 16-in. centers, 3/8-in. decorated gypsum board.
3. Heavy masonry consisting of 4-in. face brick, 4-in. cinder aggregate blocks (15 × 4 × 8-in.), 20 lb building paper, 21/32-in. square furring strips, 3/8-in. decorated gypsum board.

All windows were single glass with nominal areas of 15, 25, and 38 percent of gross wall area. Window shading consisted of awnings, venetian blinds or no shading at all. Compass orientations were: front of house facing West, or front of house facing North. Houses were either one or two story.

The floor consisted of 2 × 8-in. joists on 16-in. centers with 25/32-in. sub-flooring, 35 lb building paper and 13/16-in. hardwood finishing. The roof consisted of 2 × 6-in. rafters on 16-in. centers, 25/32-in. wood, 55 lb building paper, and triple asphalt shingles. The ceiling consisted of 3/8-in. decorated gypsum board, 2 × 6-in. joists on 16-in. centers, and 4-in. blanket insulation. Interior partitions consisted of 3/8-in. decorated gypsum board on each side of 2 × 4-in. studs on 16-in. centers.

The single story house was 48 ft across the front, and had a floor area of 1308 sq ft and a volume of 10,400 cu ft. The two-story house was very nearly square, had a floor area of 654 sq ft on each story, and also had a volume of 10,800 cu ft. For each house approximately 100 sq ft of wall area was protected from solar radiation by an attached garage. Shading due to eaves was negligible and was neglected. Each house had a basement and an attic.

DISCUSSION

C. F. KAYAN, New York, N. Y. (WRITTEN): It is indeed encouraging to see further development of some of the past work on thermal circuits and electrical simulation techniques presented at various times before the Society[1, 2, 3] during the past dozen years. This paper represents real progress, and the authors are to be commended heartily for contributing to it. Previous papers presented to the Society

[1] Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschkis (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 106).
[2] Effects of Floor Slab on Building Structure Temperatures and Heat Flow, by C. F. Kayan (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, p. 377).
[3] Electrical Analogger Studies on Panels with Imbedded Tubes, by C. F. Kayan (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 205).

on building heat flow by electrical analogue analysis have dealt with separate elements of construction such as building structure, walls under transient conditions and steady-state conditions,[2] and radiant panels with embedded pipes under steady state.[3] It is particularly interesting that here we deal with a complete *house complex*.

The thought of comparison between calculation techniques, both mathematical and by *computer*-systems, with actual physical performance under operating conditions is a most excellent and progressive idea, heartily to be commended and to be further encouraged for the future. It is only by such comparisons that ultimately our computational methods will be advanced to a successful level. In the present case, in order to make this comparison most effective, it would be helpful if the authors would provide the basic information concerning heat flow conductances and resistances entering into their parallel solution by analogue computer, as well as further technical detail on the procedure in carrying out the evaluation for one particular case analyzed.

R. G. VANDERWEIL, Boston, Mass.: I should like to ask the authors whether they found a simple relationship between actual cooling requirements and theoretical cooling load computations. According to the experience gained on test homes, can we assume that we can safely cut—say—15 percent from the cooling load computed according to THE GUIDE? If so, could we still consider the installation as one providing proper comfort? I have heard a number of discussions on this subject lately, between engineers and manufacturers, and it would be interesting to see what an actual test on 16 homes would indicate regarding possible reduction of installed refrigeration capacity compared to that computed.

C. S. LEOPOLD, Philadelphia, Pa.: As you know the word comfort has never been defined in our work nor do we have a formal definition in the TAC on Sensations of Comfort.

Not long ago I proposed somewhat facetiously the condition that people would want if they did not have to pay for the result and did not know that a test was being conducted. I think it is important when discussing conditions that people will *merely* accept, to take cognizance of what they will eventually *want* regardless of what they pay for it.

S. F. GILMAN, Syracuse, N. Y.: I am very pleased to see this type of information made available and also that more and more investigators are working on the very important problem of thermal storage.

The authors state that their thermal storage factors can be used with any cooling load calculation method. However, a previous discussor stated that there is no standard method and advocated that a method acceptable as a standard be developed. Were the results in this paper correlated with THE GUIDE method? If not, with which load calculation procedure?

In setting up the analogue, the four exterior walls were replaced by a composite wall, and the procedure is said to be justifiable on a theoretical basis. However, I doubt that it can be so justified unless a simplifying assumption is made; namely, that the radiant energy interchange between walls is negligible. If radiation effects are rigorously considered, I do not see how the four exposed walls can be replaced by a single equivalent wall, because the radiation angle factors introduce complex interdependent boundary conditions at the inside surfaces of the walls. If this simplification could actually be made, the work involved in the thermal storage research we are presently conducting would be reduced considerably. We are using the thermal circuit technique proposed by Nottage and Parmelee in a recent A.S.H.V.E. paper.* With this method, the equivalent electrical circuit is set up and then solved analytically. For the usual residential problem, if the four walls could be replaced by a single wall, the number of simultaneous equations to be solved would decrease from 36 to 18, which represents a very considerable reduction in the time required to

* A.S.H.V.E. RESEARCH REPORT No. 1497—Circuit Analysis Applied to Load Estimating, by H. B. Nottage and G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 60, 1954, p. 59).

obtain a solution. Further discussion of the composite wall hypothesis by the authors is most desirable.

I note that the outdoor air temperature was taken as a sine curve. It should be pointed out that other investigators have shown that a Fourier series representation is required for good heat flow accuracy, and that three harmonics are ordinarily necessary for north and south walls and five harmonics for east and west walls. Since this analogue utilizes sine wave generators, it would be an easy matter to consider harmonics other than the fundamental. Work along this line is suggested as a next step.

One important fact has not been made clear either in the paper or the presentation. According to the equation for determining air conditioning capacity (appearing in the section, Heat Storage Allowance for Cooling Load Calculation), if the room air temperature is maintained constant the equipment capacity required is equal to the calculated heat gain. This means there is no thermal storage unless the room temperature varies. Therefore, if my interpretation is correct, this analogue has not considered the thermal storage in the exterior walls and roof of the structure that results from the periodic variation in outdoor temperature. The given value of 13 percent of the conduction and infiltration load, which at first appeared to me to be an extremely small amount of credit for thermal storage, looks reasonable if it actually accounts only for the thermal storage in the interior portions of the residences. This point needs clarification.

A. B. NEWTON, Wichita, Kan.: I have a question about Fig. 1. Is it true that we indicate that 99 percent of the people were happy or desired a temperature of about 85 F as a maximum temperature for comfort? I would appreciate the authors' comment on this chart.

AUTHORS' CLOSURE (T. N. Willcox): We appreciate Professor Kayan's kind remarks on the contribution made by this paper. The need for a more detailed explanation covering the various parts of the thermal circuit is recognized. In the interests of showing the overall picture covering a rather voluminous study some worthwhile subjects were drastically compressed. It is our intention to prepare future papers on analogue computer analyses and go into more detail on specific phases.

Dr. Gilman has asked by what heat gain load calculation method the results were obtained for correlation with thermal storage factors. The method used was the one which we developed several years ago. The method recently recommended by the *National Warm Air Heating and Air Conditioning Association* is basically the same in procedure and end results as the authors' method used before the completion of these analogue computer studies. The *A.R.I.* method or any similar method giving basically the same end results would be similarly satisfactory with a reasonable storage allowance factor consistent with the method used and providing it is based on instantaneous peak load or a peak load averaged over a short time such as 3 or 4 hr. The actual heat storage allowance will be changed slightly depending on the heat gain method used. The paper illustrates how to find the magnitude of the heat storage effect and correlate it with any acceptable heat gain calculation method. Three simplifications questioned by Dr. Gilman were: (1) A single composite outside wall replacing the four individual outside walls; (2) omission of reradiation and the resultant interaction of one internal wall temperature on another; (3) the use of a sine curve for periodic outdoor temperature variations. These simplifications were justified on the basis of a comparison of computer load results with published test results from the University of Illinois as explained in the paper and other carefully correlated studies. Dr. Gilman points out one of the compelling reasons for such simplification—the practical necessity for reducing the bulk of simultaneous equations to be solved on a computer of limited size.

One further point raised by Dr. Gilman is the interpretation of the equation:

$$Q_o = Q_i - (C_1 + C_a)\,(dT_a/dt)$$

This shows that equipment capacity must equal room heat gain so that room temperature does not rise. Room heat gain is only a portion of the heat absorbed by the exterior surfaces of a house. Some of the heat entering exterior surfaces is stored in the walls and given back outdoors at a later time. Likewise some is given to the room after a time delay. This time delay effect on the outer walls is very carefully taken into account in the thermal circuits and is explained in some detail in the section on the thermal circuit.

Mr. Leopold's comment on the definition of *comfort* brings out an interesting phase of this work. In order to determine from personal interviews the indoor temperatures acceptable to users based on their experiences in living with air conditioning they were asked *What is the highest acceptable indoor temperature in the summertime?* and *What is the lowest acceptable, etc?* The questions indicate that *comfort* is relative over a range of temperatures and that owners will accept some variation. Answers were, of necessity, based on the indoor dry-bulb temperature shown by thermostat thermometer and on individual preferences. Plotting the numerical answers to these questions gave the results shown in Fig. 1.

To Mr. Newton's question on the meaning of Fig. 1, we can answer that the points on the upper curve give maximum indoor temperatures and percentages of the sample users who would accept that maximum temperature *or a lesser* temperature. For the point mentioned and with a true probability curve only one percent of the users would accept an indoor temperature higher than 85 F.

Mr. Vanderweil's comment requests a simple relationship between calculated load and actual. This paper gives for residences a heat storage allowance which is simple to apply. The method has been successfully used on a substantial number of homes of various architectural designs and construction materials. Future advances in the art may further simplify the utilization of heat storage for cooling load reduction.

AUTHORS' CLOSURE (C. T. Oergel): The wall has been represented by a π circuit, three resistances and two capacitances, for which there will be two first order differential equations. When solving a heat flow problem, it is first necessary to determine the temperatures. The wall equations should be written in terms of temperature and the temperature of the capacitance closest to the indoor house temperature determined. This will be a second order differential equation, and there will be one for each one of the four walls.

These equations are now to be converted to heat flow equations, *i.e.*, differential equations which determine heat flow instead of temperature. The heat flow per square foot of wall is written in terms of conductance, capacitance to the air in the house, and the temperature difference. This expression is then used to convert the temperature equation to the heat flow equation. Now the four walls are alike, at least they were assumed to be so in the houses we calculated. Therefore, the lefthand sides of the differential equations for the four walls will be identical. Only the imposed or impressed functions on the right-hand sides of the equations will differ, particularly the sun loads.

Each wall equation is multiplied by the respective wall area and the four equations are added. The form of the left-hand side will remain unchanged except that it contains total heat flow while the right-hand sides will add to give the total impressed loads. This equation shows that the total wall heat flow can be reproduced by applying the total sun load to a composite wall, a single π circuit.

This equivalence of composite wall and the four separate walls is only true for heat flow. It is not true for temperatures. It is, therefore, not possible to probe into this composite wall circuit to determine temperatures. This is easily seen. I think it is rather obvious, because with the sun shining on the West wall but not on the East wall, it is certainly clear that the outside wall temperatures and hence temperatures throughout the two walls, will differ. We have used the composite wall only for heat flow measurements.

1520

No. 1520

THE OPERATING COST OF RESIDENTIAL COOLING EQUIPMENT

By S. F. GILMAN,* L. A. HALL** AND
E. P. PALMATIER,† SYRACUSE, N. Y.

ALTHOUGH satisfactory methods for predicting the operating cost of residential heating systems[1,2,3] have been available for several years, an authoritative method for predicting the cost of operation of residential cooling equipment does not yet exist. The rapidly increasing demand for residential air conditioning during the past few years makes it imperative that a method be developed which will gain acceptance by the air conditioning industry. With such a method, a potential customer could be given a reasonably accurate prediction of the monthly or seasonal cost of operating his contemplated equipment.

The objectives, therefore, of this investigation were:

1. Development of a method for predicting the operating cost of residential cooling equipment.

2. Investigation of the validity and generality of the method by collecting and analyzing operating data from several residences in different parts of the country.

DERIVATION OF METHOD

Over any period of time the power cost S of residential cooling equipment. expressed in dollars, can be predicted from the equation

$$S = RPT/100 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

R = power rate, cents per kilowatt hour.
P = power input to equipment, kilowatts.
T = predicted operating time of equipment, hours.

Paralleling the degree-day method of estimating heating costs, the hypothesis is made that the amount of heat removed from a residence during cooling is

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† Director of Research, Carrier Corp. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June 1954.

proportional to the number of degree-days above a datum temperature t_d. In equation form,

$$CT = KD \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

C = total cooling capacity (sensible plus latent) of equipment, Btu per hour.
D = number of degree-days above the datum temperature t_d.
K = a constant of proportionality.

The value of D can be computed from available Weather Bureau data and the equation

$$D = \sum_{i=1}^{n} (t_a - t_d)_i \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

t_a = average outdoor dry-bulb temperature for each individual day, Fahrenheit.
t_d = temperature base for computing degree-days, Fahrenheit.

Therefore, D is evaluated by subtracting the datum temperature from the average temperature for each day and then summing up over the period under consideration. It is important to note that negative degree-days are not considered; hence, Equation 3 is restricted to the condition that $t_a \geq t_d$.

The constant of proportionality K can be evaluated from the conditions on a design day. On such a day, Equation 3 reduces to

$$D = t_m - t_d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where t_m is used to distinguish between the average temperature for any day, t_a, and that for a day on which the design outdoor conditions are fulfilled, t_m. It should be noted that t_m is the design outdoor dry-bulb temperature minus one-half the daily range and, as a consequence, is easily determined for most localities from available publications.[4,5] Furthermore, the amount of heat removed on a design day can be expressed as:

$$CT = 24\,H \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where H is the average hourly cooling load over the 24-hr period. Thus by substituting Equations 4 and 5 in Equation 2

$$K = (24\,H)/(t_m - t_d) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The generalized form of Equation 2 is therefore

$$CT = (24\,HD)/(t_m - t_d) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

or in terms of predicted operating hours,

$$T = [24\,HD]/[C(t_m - t_d)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The principal unknown in this equation is t_d. Once its value is determined, the operating time T can be readily calculated and used in Equation 1 to predict the power cost. Consequently, an important phase of this investigation concerned the evaluation of the datum temperature t_d.

It should be emphasized that the value of H to be used in Equation 8 is not the maximum instantaneous heat gain but the *average hourly cooling load over*

the 24 hr comprising the design day. At least one cooling load calculation method which seeks to evaluate H has been reported.[6,7] Therefore, for at least this method, the symbol H can be interpreted as the *calculated cooling load.* There are undoubtedly other load calculation methods which also seek to evaluate H but which do not appear in the technical literature. For the purposes of this investigation, H has been determined by the method reported.[6]

APPARATUS

Of the eleven residences used during this investigation, data were collected from two during the 1952 cooling season and from the remaining nine during

TABLE 1—DETAILS OF RESIDENCES TESTED

Residence No.	City	Description	Floor Area, sq ft	Nominal Unit Capacity, Tons	Condensing Method	Cooling Load, Btu/hr	Load Factor*	Duration of Test
1	Atlanta	Tan Brick Ranch Style	2984	5	Cooling Tower	49,200	0.82	5/20 - 9/1/53
2	Atlanta	Brown and Grey Cape Cod	2450	2	Air	22,760	0.95	6/20 - 10/1/53
3	Washington	White Brick Virginian	1800	3	Waste Water	36,180	1.01	6/17 - 9/8/53
4	Washington	White Brick Garrison	2100	5	Cooling Tower	38,830	0.65	6/17 - 9/8/53
5	Houston	Red Brick Ranch Style	2424	3	Cooling Tower	38,000	1.05	7/9 - 7/22/53
6	Dallas	Tan Brick Ranch Style	1640	3	Cooling Tower	32,150	0.89	7/30 - 8/21/52
7	Dallas	Pink Brick Modified Ranch	3165	5	Cooling Tower	37,750	0.63	6/3 - 10/5/53
8	Dallas	Red Brick Ranch Style	2248	5	Cooling Tower	29,470	0.49	6/3 - 10/5/53
9	New Orleans	Red Brick Colonial	3365	3	Cooling Tower	34,000	0.94	7/12 - 8/13/52
10	New Orleans	Red Brick Colonial	3160	5	Cooling Tower	51,940	0.87	7/10 - 11/2/53
11	Syracuse	Green Ranch Style	967	2	Cooling Tower	21,400	0.89	6/19 - 9/1/53

* Ratio of cooling load to nominal unit capacity.

the 1953 season. These residences, which were made available through the cooperation of the owners, were occupied in their usual manner except that the occupants were requested to keep all windows and outside doors normally closed and to maintain a thermostat setting between 72 F and 75 F. The locations of the residences and other pertinent details are given in Table 1. It should be noted that the floor areas range from 967 to 3365 sq ft and the calculated cooling loads range from 21,400 to 51,940 Btuh. Since three different condensing methods are also cited, the ten residences and air conditioning systems evidently have many dissimilarities. A view of Residence 7 is shown in Fig. 1.

A separate watthour meter installed in the air conditioning circuit measured the total power consumption. The number of hours of compressor operation were obtained with a timer wired into the compressor circuit. In addition,

Residences 1 to 6, inclusive, and 9 were equipped with chart-type half-hour demand meters.

Additional instruments used during the calibration of each air conditioning unit were the following: revolution counter and stop watch, sling psychrometer, wattmeter, manometer and orifices, laboratory-type pressure gages, precision potentiometer and copper-constantan thermocouples.

PROCEDURE

Before beginning the regular collection of operating data, each residence was visited by Research Department personnel and the air conditioning equipment calibrated. The purpose of the calibration was to obtain a precise evaluation of the actual cooling capacity C and total power input P as a function of the outdoor wet-bulb or dry-bulb temperature, depending on the condensing method.

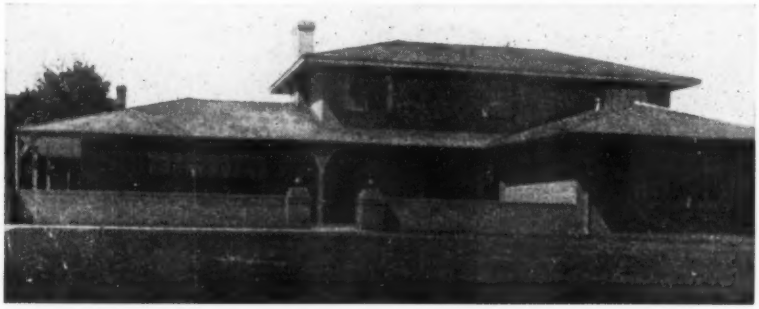


FIG. 1. VIEW OF RESIDENCE 7 (DALLAS), LOOKING NORTH

Representative results for units equipped with cooling towers are presented in Fig. 2. For the waste water applications the capacity and power input were considered constant throughout the season.

While calibrating the 11 units, the orientation of each residence was determined and several photographs taken. In addition, the effects of shading due to trees, adjacent structures, etc., were noted, and a set of blueprints or sketches procured, together with construction details. With this information, the cooling load H was calculated* by the so-called *24-hr estimating method*.[6]

During the cooling season, readings of the watthour meters and timers were taken periodically by field personnel and forwarded to the Research Department in Syracuse. From these readings, the actual operating hours and total power consumption of each unit were obtained over periods of approximately one month's duration. In addition, for seven of the residences more detailed data were secured from the demand meter charts, one of which is illustrated in Fig. 3. In conjunction with weather data, the charts were used to determine the operating time and cooling capacity for each 24-hr period.

Weather data for each city were procured from *Local Climatological Data* sheets issued monthly by the U. S. Weather Bureau. These data were used to obtain appropriate values of daily average dry-bulb temperature t_a for use in

* Sample copies of the load calculation form can be secured upon request to the authors.

Equation 3, and also to determine values of C and P from graphs typified by Fig. 2.

Data for evaluating the mean temperature, t_m, which is the design outdoor dry-bulb minus one-half the daily range, were obtained from available material.[4,5]

With the data collected over test periods of about one month's duration, t_d was evaluated from Equation 8. Since t_d also appears in the definition of D given by Equation 3, the solution was obtained by a trial-and-error procedure.

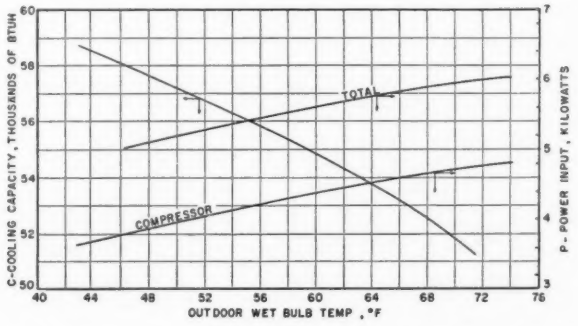


FIG. 2. EQUIPMENT CALIBRATION CURVES FOR RESIDENCE 1 (ATLANTA)

The data collected from the demand meters also made possible an analysis of the operating data from seven of the residences on a daily basis, as follows: For any one day, Equation 3 reduces to

$$D = t_a - t_d \quad \ldots \ldots \ldots \ldots \ldots \quad (9)$$

and Equation 8 becomes

$$T = [24\,H\,(t_a - t_d)]/[C(t_m - t_d)] \quad \ldots \ldots \ldots \quad (10)$$

Solving for t_a,

$$t_a = t_d + (t_m - t_d)\,(TC/24\,H) \quad \ldots \ldots \ldots \quad (11)$$

Dividing by t_m, there is obtained

$$\frac{t_a}{t_m} = \frac{t_d}{t_m} + \left(1 - \frac{t_d}{t_m}\right)\frac{TC}{24H} \quad \ldots \ldots \ldots \quad (12)$$

which is dimensionless and therefore a generalized equation for correlating all data obtained on a daily basis. The form of the equation shows that t_a/t_m is a linear function of the variable $TC/24H$. Therefore, the data from each residence were plotted in the manner illustrated by Fig. 4, where the curve has been determined by the application of the method of least squares[8] to the plotted points. The intercept of the curve yields the value of t_d/t_m, from which t_d can be determined. To minimize heat carry-over effects, the period from 6:00 a.m. to 6.00 a.m. the next day was considered as a day when compiling the data. By this procedure, values of the datum temperature t_d were obtained from daily operating data gathered in seven of the eleven residences.

RESULTS AND CORRELATION

Representative results obtained from the daily operating data are presented in Fig. 4, where each plotted point represents a particular day. These daily data show considerable spread, which is partially attributable to thermal storage in the structure and contents resulting from the periodic variation of the outdoor temperature. The fact that the temperature at 6:00 a.m. on one day is usually different from that at 6:00 a.m. the following day also contributes to the spread. The points correspond to various values of percent sunshine, wind velocity, and some include the influence of rain. An analysis of the effects of sunshine and wind did not disclose any trend in the results due to these variables; hence, they

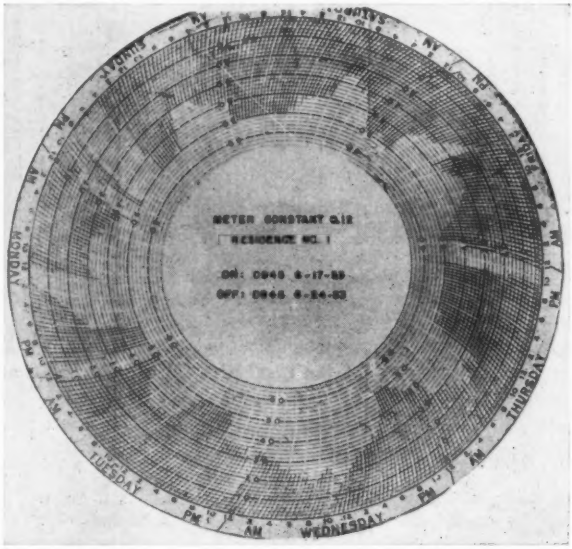


FIG. 3. TYPICAL DEMAND-METER CHART FROM RESIDENCE 1 (ATLANTA)

were not primarily responsible for the scatter. The intercept of the curve has a value of 0.825, which when multiplied by the t_m value of 86 for Atlanta yields a datum temperature of 70.9 F.

The results for the 11 residences are presented in Table 2. The difference between the test periods in Column B and Column D is due to the fact that the analysis made on a daily basis permitted screening of the data and discarding those which were not valid. For example, if the occupants of a residence inadvertently turned the unit off before leaving for a week-end, the daily record would show this but the monthly record would not.

Comparison of the two datum temperatures obtained for seven of the residences shows close agreement with the exception of Residence 3, for which the Column C value is 5.4 deg lower than that in Column E. Re-examination of the

test data did not disclose a reason for this apparent inconsistency in the results. Since the Column E value of t_d is in good agreement with others in that column, it is most probable that the period used to obtain the Column C value included some unknown effects which cause the inconsistency.

Another inconsistency exists in the value of t_d for Residence 8, because the value of 57.0 F is 13.7 deg below the 70.7 F weighted average for the remaining ten. Referring to Equation 8, this apparently erroneous value could result from the calculated cooling load H being less than the actual load. An analysis of the structure, air distribution system, and load estimating form[6] indicated the value of H was indeed too low. This residence had much of the supply-air ductwork in the attic and the return-air ductwork in a crawl space and, as a consequence, the duct heat gain was considerable. The load estimating form[6] is presently being refined to more accurately evaluate heat gains to ductwork

TABLE 2—RESULTS OF DATUM TEMPERATURE EVALUATIONS

RESIDENCE No.	FROM DAILY DATA		FROM MONTHLY DATA	
	Test Period, days	t_d F	Test Period, days	t_d F
A	B	C	D	E
1	61	70.9	103	71.2
2	17	67.7	102	67.5
3	21	66.4	48	71.8
4	76	70.1	70	72.0
5	12	69.2	14	69.2
6	12	69.4	13	71.9
7	—	—	126	71.5
8	—	—	153	57.0
9	24	69.4	31	72.0
10	—	—	114	71.5
11	—	—	74	68.8

in attics, basements, and crawl spaces, and thereby yield a more precise evaluation of H.

The results in Table 2 show that t_d varies somewhat from residence to residence. The cooling load of a residence is comprised of infiltration (or ventilation), transmission, solar, and internal components. When $t_a = t_d$, the net load over a 24-hr period will be zero. Since the solar and internal components represent heat gains, the infiltration and transmission components must be heat losses. The datum temperature will therefore be lower than the room temperature. Since the relative magnitudes of these load components will vary from residence to residence, the datum temperature can also be expected to be a variable. However, the results in Table 2 show the variation in datum temperature is small. Consequently, it is concluded that t_d can be considered *constant* with sufficient accuracy for all practical purposes and, furthermore, *its value should be 70 F.*

A June 1948 study of the temperature conditions in several normally occupied residences has been reported.[9] When the residences were neither heated nor cooled, it was found that the daily mean indoor temperature remained 5 or 6 deg above the daily mean outdoor temperature over a range of weather conditions. Taking 75 F as the design indoor condition the predicted datum temperature would be 5 or 6 deg less, hence 69 or 70 F, which corroborates the conclusion drawn from this investigation.

As additional correlation, data secured from an A.S.H.V.E. paper[10] reporting a 48-day test of a residence at the University of Illinois yielded 71.1 F for t_d. Since the test was conducted with no internal load, the datum temperature would be slightly higher than that for a normally occupied residence. Taking this into account, the value correlates well with the 70 F result obtained from this investigation. Moreover, in a report[11] of tests conducted in Kansas City, a tentative conclusion is drawn that the power consumption of residential cooling equipment is related to degree-days above 70 F.

For the 11 residences, the actual operating times and those which would be predicted on the basis of a 70 F datum temperature are given in Columns 3 and 4 of Table 3. Neglecting Residence 8, the results for which have previously been stated to be erroneous, the deviation between the actual and predicted hours of operation ranges from −19.3 to +9.6 percent, a rather low order of accuracy. However, the difference in dollars of power cost, Column 9, ranges

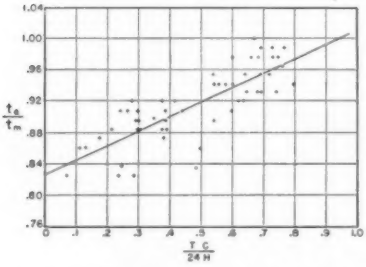


FIG. 4. RESULTS OF ANALYSIS OF DAILY DATA FROM RESIDENCE 1 (ATLANTA)

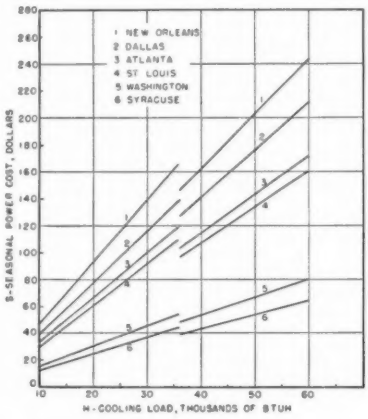


FIG. 5 *(right)*. TYPICAL PREDICTED POWER COSTS FOR SIX CITIES

only from −$5.27 to +$14.25 for the stated periods. Note that although the operating hours for Residence 11 deviate by 19.3 percent, the difference between the actual and predicted power cost is less than $5.50 over a 10-week period. This residence is located in Syracuse, which has a short, cool summer. Hence, the operating hours predicted for northern regions of the country may be considerably in error, but the predicted dollars of power cost, which is of primary concern to the owner, may be within a few dollars of the actual.

The order of accuracy of the method when applied to various regions can be analyzed through inspection of Equations 3 and 8. In Equation 8, the larger t_m, the less is the error involved in T by accepting a single value of t_d, because deviations in t_d have less influence on the term $(t_m - t_d)$ when t_m is large. Recalling the definition of t_m; namely, design outdoor dry-bulb temperature minus one-half the daily range, better accuracy can evidently be expected in the hotter regions having small daily ranges. Examples would be New Orleans, Dallas, and Kansas City. Referring now to Equation 3, it is also evident that the larger t_a, the less is the error involved in D as a result of an approximate

value of t_d being used. Therefore, the prediction of T should be most accurate in the regions of the country having long, hot summers.

On the basis of this analysis and the test results, it is concluded that acceptance of a single value of 70 F for t_d will ordinarily yield the predicted power cost for a normal cooling season within ± 8 percent or \$10.00, whichever is greater. Such an order of accuracy is deemed to be satisfactory; indeed, greater accuracy can hardly be justified when it is realized that the year-to-year variation in the total number of degree-days accumulated will frequently be considerable. It must be recognized that for certain residences, especially those of unusual design, the accuracy of the prediction will sometimes be outside the stated range. If, however, the 24-hr average cooling load H is carefully evaluated, instances of poor accuracy predictions will be rare.

For regions of the country having a hot, dry climate, an analysis of the components comprising the cooling load indicates the optimum datum tempera-

TABLE 3—COMPARISON OF ACTUAL AND PREDICTED OPERATION

Residence No.	Duration of Test, days	Actual Operating Time, hr	Predicted Operating Time, hr	Deviation, Referred To Actual, Percent	Power Input, kw	Power Cost*		Difference, Dollars
						Actual	Predicted	
1	2	3	4	5	6	7	8	9
1	103	1270	1307	+ 2.9	5.95	\$151.13	\$155.33	\$+ 4.20
2	102	1202	1107	− 8.0	2.35	56.49	51.96	− 4.53
3	48	437	479	+ 9.6	3.50	30.59	33.53	+ 2.94
4	70	436	476	+ 9.2	6.10	53.19	58.07	+ 4.88
5	14	294	278	− 5.4	4.70	27.64	26.14	− 1.50
6	13	292	301	+ 3.0	4.50	26.28	27.09	+ 0.81
7	126	1326	1403	+ 5.8	6.30	167.08	176.78	+ 9.70
8	153	1611	1285	−19.3	6.20	199.76	159.34	−40.40
9	31	388	417	+ 7.4	4.10	31.82	34.19	+ 2.37
10	114	1253	1360	+ 8.5	6.66	166.90	181.15	+14.25
11	74	459	370	−19.3	2.93	26.95	21.68	− 5.27

* Based on 2 cents per kwhr rate

ture may be a few degrees below 70 F. This results from the increased solar radiation which is characteristic of such a climate. Although it is believed the datum of 70 F is sufficiently accurate for these areas, correlation of operating data from hot, dry regions may be desirable.

With a value of 70 F for t_d in Equations 3 and 8, the seasonal power cost curves of Fig. 5 were predicted from Equation 1. The degree-days used in the computations were average values over a period of several years, and the power rate was that currently charged in the locality. The higher curve for New Orleans as compared with Dallas is primarily a result of the higher power rate in the former. The break in each curve at 36,000 Btuh cooling load is merely the result of the lesser kw/Btu requirement of this particular 5-ton unit.

The purpose of Fig. 5 is to provide an insight into the approximate power costs to be expected with units utilizing cooling towers. Since the curves are based on power input and capacity data applicable to a particular manufacturer's equipment, they *must not be construed* as working curves for predicting power costs. Instead, the following procedure should be employed for each application

of a manufacturer's equipment and for each type of installation; *i.e.*, air-cooled, cooling tower or waste water.

PRACTICAL UTILIZATION OF METHOD

To facilitate application of the method, the charts in Figs. 6 and 7 have been formulated. The first is used to predict the number of hours the equipment will operate and the second to obtain the resultant power cost. For convenience and greater accuracy in the low ranges, a second scale has been provided along the bottom of each chart.

The average capacity C and power input P over a month or season in a particular locality must be obtained from the equipment manufacturer. The 24-hr

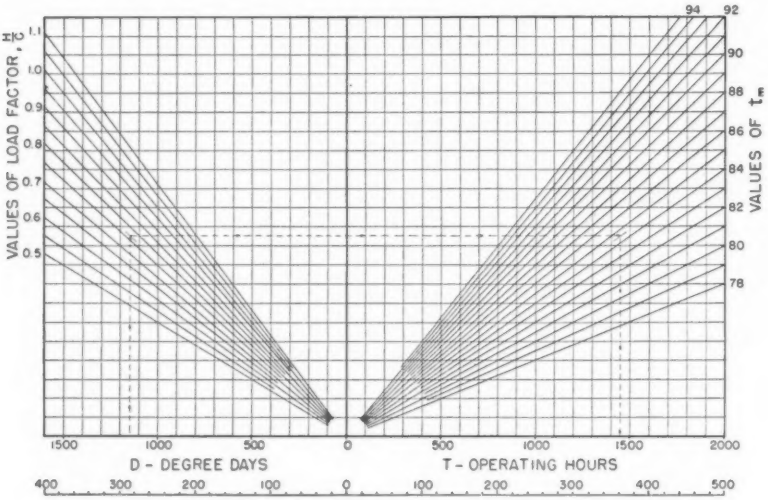


FIG. 6. CHART FOR PREDICTING OPERATING HOURS

average cooling load at design conditions H should be obtained by the published method,[6] or an equivalent one. Tabulated degree-days above 70 F are not presently available in the literature, but it is hoped the U. S. Weather Bureau will begin listing daily values in the *Local Climatological Data* sheets which are issued monthly for numerous localities. For the present, values can be obtained from available degree-day maps.[12,13] Alternatively, fairly accurate values can be readily procured from Weather Bureau data by subtracting 70 from the average monthly temperature and multiplying by the number of days in the month. By doing this for each summer month over a period of several years, values fairly representative of normal summer weather will be obtained. The hotter the weather, the better will be the agreement with the result of computations based on the daily average temperatures. It is recommended, however, that the TAC on Weather Data consider sponsoring a research project to secure more precise average values from weather data collected during the past several years.

The following example illustrates the use of the charts:

Given:

Total degree-days for season:	1150
Design dry-bulb temperature:	95 F
Daily range of dry-bulb:	21 F
24-hr average cooling load:	28,500 Btuh
Seasonal average capacity of equipment:	37,000 Btuh
Seasonal average power input:	4.5 kw
Power rate:	2 cents per kwhr

Solution: The load factor H/C is 28,500/37,000, or 0.77. Half of the daily range is 10.5 deg, which when subtracted from the design dry-bulb of 95 F yields 84.5 F for t_m. Then, from Fig. 6, 1450 is the number of predicted operating hours. Using

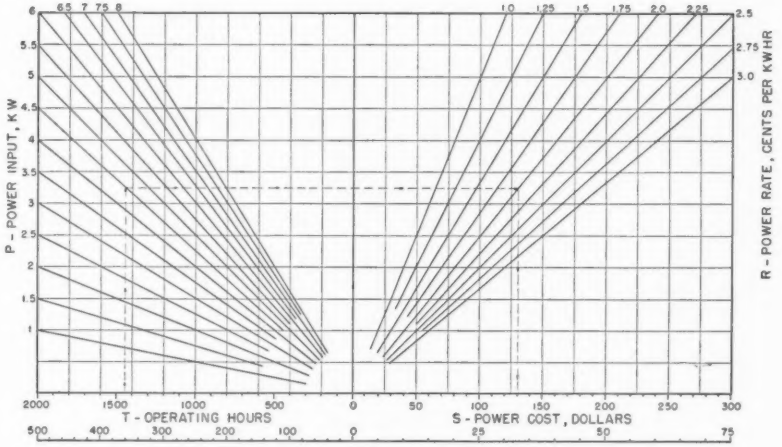


FIG. 7. CHART FOR PREDICTING POWER COST

this value of T in Fig. 7, the predicted power cost is $130.00. For an air condensing application this also represents the total operating cost. For a water condensing application the water cost, as determined from the operating hours, water rate and manufacturer's data on equipment water usage, is added to the power cost to obtain the predicted total operating cost.

CONCLUSIONS

The following conclusions were drawn from this investigation:

1. The operating cost of residential cooling equipment is proportional to degree-days above 70 F.
2. Since the procedure for calculating cooling degree-days is similar to that for calculating heating degree-days, the *degree-day* concept is preferable to the *degree-hour* concept frequently postulated for cooling.
3. Although both the analysis and test results indicate the datum temperature varies somewhat from residence to residence, for all practical purposes 70 F can be used irrespective of the residence and its location.

4. It is desirable that the Weather Bureau add cooling degree-days above 70 F to the *Local Climatological Data* pamphlets which presently list heating degree-days below 65 F.

5. The method which has been developed and validated will ordinarily enable prediction of the operating cost of residential cooling equipment during a normal season to within ±8 percent or $10.00, whichever is greater.

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DISCUSSION

H. T. GILKEY and D. R. BAHNFLETH, Urbana, Ill. (WRITTEN): This paper represents the first step in the solution of a problem which has become very important in the field of residential air conditioning. Although the final results present a consistent picture of datum temperature and operating cost, it does not seem that a study of only 11 residences can be used to establish the datum temperature for estimating residential cooling costs. It may well be that future work will show that the results of this study are entirely reasonable. Other work must be done, however, and many more residences must be studied before this datum can be definitely established.

A very interesting aspect of the problems encountered in residential air conditioning is shown in Fig. 2. It may be noted from Table 1 that the nominal cooling unit capacity of the equipment in Residence 1 is 5 tons. Yet the information in Fig. 2 indicates that the outdoor wet-bulb temperature would have to drop below 40 F before the cooling unit capacity would reach this value. Furthermore, in Atlanta, Ga., the location of the residence, the outdoor design conditions are 98 F dry-bulb and 76 F wet-bulb temperatures. With a 76 F wet-bulb temperature, the capacity of the cooling unit would be approximately 4 tons of refrigeration instead of the rated

5 tons. It should be mentioned that the performance of this equipment is typical, and that even when water conservation methods are not applied, it has been found that the actual cooling unit capacity is usually below the rated capacity. This situation causes one to wonder if the rating methods are not misleading as well as unrealistic.

The statement is made in the paper that when t_a, the mean temperature for a day, equals t_d, the datum temperature, the net load over a 24-hr period will be zero. The paper then continues that since the solar and internal components represent heat gains, the infiltration and transmission components must be heat losses. This is a little mystifying since usually the cooling unit at some time during the day removes some heat. Perhaps one is to infer that this statement applies only to a residence which is not being cooled by refrigeration. It is agreed, however, that the outdoor temperature is only one of the factors which contribute to the cooling load of a residence. Because it is felt desirable to use outdoor temperature as a basis for predicting operating costs, it is reasonable that the datum temperature would be lower than the control temperature within the residence.

A reference is made in the paper to a 48-day test conducted at the University of Illinois. The authors state that the datum temperature for this study was 71.1 F. Apparently, the authors have not had complete information since a preliminary analysis of the data indicates that the datum temperature should be in the vicinity of 65 F for this study. It is hoped that within the near future more complete information will be available on this phase of the work at the University, and that the period covered in the report can be longer than the 48 days mentioned in this paper.

One of the questions which occurs concerning the results of this study concerns the weather data used to determine the predicted values shown in Table 3. In order to get the good agreement shown in the table, it is probable that the weather experienced during the test period for each house was used to obtain the predicted operating time and power cost. One then wonders how the weather during this period varied from so-called seasonal weather for the same localities. Before the authors' assurance that the predicted costs will be within ± 8 percent or $10.00, whichever is greater, can carry much weight for the home owner, it would seem that a *normal* season should be defined for each locality.

It is hoped that the authors and others will continue this very important work. Much more information is needed so that all variables such as climate, type of construction, method of cooling the condenser, etc., can be considered. In addition, attention should be given to the development and publication in THE GUIDE of a method of residential cooling load calculation. This move would permit future work in residential air conditioning to be reported on a uniform basis.

R. C. CHEWNING, Portland, Ore. (WRITTEN): The effort to devise a method of predicting residential cooling costs reported in this paper is very timely in view of the increasing volume of residential cooling. The problem involved is vital to a large section of industry and is one in which the answers required are of grave concern to the public in order that they can properly evaluate the cost of owning and operating summer cooling systems which they might install.

Unfortunately, this discussion of the paper must be based on experience and tests with air conditioning systems in commercial buildings rather than residences. The types of buildings included in this experience are a 12-story office building with a very large percent of the exterior glass, a 6-story 200 ft square building used as a newspaper office and publishing plant, and a bank building 50×200 with 4 floors plus a basement. These buildings are mentioned to illustrate the types of buildings with which the experience has been gained.

The correlation of data from flow meters and temperature recorders on the system serving the 12-story office building, with data on outside temperature indicated the only basis on which a relationship between heating requirements and outside temperature could be obtained was on a 24-hr average temperature basis.

No attempt was made to correlate cooling requirements with outside temperature, but indications were that the same principle would apply. This is in agreement with the authors' findings.

However, the experience with commercial installations would indicate that 70 F would be too high a value of t_d as defined in the paper. This is not a contradiction of the authors' findings, but rather a warning that use of the data developed in paper should be restricted to residential applications.

Commercial installations require cooling for interior spaces even when the outside temperatures are at design heating conditions. Unless the system serving such space has been designed to use outside air for cooling the space, cooling equipment must be operated. Further, in commercial installations, the operation of cooling equipment is required at 65 F outside temperature and with 24 hr average temperatures as low as 60 F and 55 F.

One factor noted in the tests reported might indicate that even for residences the value of 70 F for t_d might be high. Nine out of eleven of the residences listed in Table I are of masonry construction, brick. This type of construction has high thermal storage and results in considerable lag between inside and outside temperatures. The result is to emphasize the 24 hr average outside temperature effect and to reduce the system operation in maintaining inside temperatures. It is possible that further tests should be made on standard frame construction before definite conclusions are made.

A. E. DIEHL,* York, Pa. (WRITTEN): The authors have done a very good job in setting up a mathematical method of arriving at operating costs for residential air conditioning based on a degree day basis.

Information of this type is necessary and becoming more important every day in view of the fact that FHA and its field agencies are requiring a statement with respect to operating costs along with an application for a loan on a residence containing an air conditioning unit.

This article should be qualified in that it deals entirely with the single compressor type of unit and therefore the results obtained are based on actual data received on a single compressor series of installations.

We raise this point because we believe that the two compressor approach will result in reduced operating costs as compared with a single compressor type unit.

The basis for our analysis is that the cooling load calculation which plays a very important part in Equation 8 which is used basically to establish operating costs is an integrated figure arrived at by calculating instantaneous figures for various hours of the day and dividing by 24.

We therefore believe that the ideal approach would be to have a compressor capacity that would balance the instantaneous type of load obtained in a residence if the maximum efficiency with respect to operation costs are to be obtained. Although the two compressor unit does not approach this condition of an unlimited number of steps of capacity available, it does have the ability to operate at 50 percent capacity for a great period of the time.

Therefore, we would suggest that this point be raised since the subject paper should not be considered as the final analysis on the method and the results obtained in calculating cooling operating costs.

If a unit is designed so that only half of the capacity is used when needed, naturally operating cost reductions are going to be secured. Unfortunately we do not have sufficient data with respect to operating times, etc., for various parts of the country and therefore we can't offer any concrete data at this time regarding two unit operation.

On the other hand, it might be advisable to suggest to the authors of this paper that they consider the inclusion of information in forms of a curve that would correct operating periods for the use of integral steps of compressor capacity for residential air conditioning.

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* Chief Engineer, Commercial Products, York Corp.

F. R. ELLENBERGER,† Bloomfield, N. J. (WRITTEN): The authors have presented an excellent paper on a subject of importance to the industry.

I would like to ask the authors if any attempt has been made to evaluate the accuracy of predicted power consumption with datum temperatures other than 70 F. For example, if t_d is taken as 75 F, the denominator of Equation 8 will be reduced—but so will the numerator, because of the corresponding reduction in degree-days, D. It is possible that a datum temperature of 75 F would still be a reasonable match for the data collected in the field program described. An inspection of Fig. 4, for example, indicates that a line of slightly less slope and with an intercept at a higher value of t_s/t_m would perhaps be a better match for the data at the lower values of $TC/24H$. This brings up the question as to the validity of the least squares method which, although providing the best overall relation for the plotted points, gives undue weight to higher values of $TC/24H$, when the objective is to establish the intercept at $TC/24H = 0$.

A few years ago my company sponsored a field test program on packaged air-to-air heat pumps,†† for which one of the objectives was to obtain accurate power consumption data. Data were automatically recorded at least once each hour and cooling data were subsequently plotted in the forms of Btu/hr *vs.* local outdoor temperature, and Kwh/hr *vs.* temperature. By this means it was possible to average data for each hour over an extended period of time covering one or more complete cooling seasons. Naturally, due to transient and cycling effects, the individual hourly points thus plotted exhibited a considerable spread, but when the kwh/hr and Btu/hr values observed at each outdoor temperature were averaged over the extended test period, they showed an excellent match to a straight line. The equation for this straight line in the case of Btu/hr *vs.* temperature is basically the same as Equation 8 of the present paper.

In all residential installations the intercept at zero capacity and power occurred very close to a temperature of 75 F. As would be expected, some hours were observed where cooling was required below 75 F, and the best match for all the data included a *hook* on the bottom of the straight line. However, the total hours of operation below 75 F were so few that seasonal averages were not materially influenced. On installations where the indoor fan was operated continuously during the summer, this *hook* was more pronounced in the kwh/hr *vs.* temperature plots. In all cases, however, no individual averaged point above 75 F varied from the straight-line average by more than 2 deg.

It was tentatively concluded from the results of these tests that the effects on operating costs of solar heat gains were largely cancelled out over a number of hours. Over short periods of time, however, solar heat gains did, of course, have a material influence on cooling requirements, hence, for establishing installed capacity could not be ignored.

If, then, when operating data are collected over long periods of time from many installations, it should be established that 75 F is a more typical datum temperature —or even if it is less valid than 70 F, but still yields operating cost estimates within reasonable accuracy—then convenient use can be made of weather data already available.** This information is tabulated in the form of cooling degree-days above 75 F, and has been compiled for 227 locations through the U. S. The tabulations cover the years of 1898 to 1940, and give cooling degree days for each month as well as for each year.

I have an additional question concerning the effect of *microclimate*, or the variations of weather conditions from one place to another within a given geographical

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† Manager, Advanced Product Engineering, Weathertron Department, General Electric Co.

†† Evaluating Heat Pump Performance, by F. R. Ellenberger, A. B. Hubbard, W. R. Foote, F. Burggraff and J. J. Martin, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 87).

** U. S. Weather Bureau Forms DD-2AC.

area. It is not uncommon to find, particularly in hilly territory, local differences in average daily temperatures that are consistently 5 deg, or even as much as 10 deg apart. Therefore, unless the test houses used in this paper all fortunately had about the same temperatures as the nearest Weather Bureau Station, the facts that led to the conclusion that a maximum error of 8 percent in predicting operating hours of cooling equipment was justified, may have been, to some extent, fortuitous. In other words, if the actual degree-days experienced at a given spot can vary more than 8 percent than the degree-days observed at the nearest weather station, it would be impossible to obtain a precision of 8 percent when Weather Bureau degree-days are used to predict operating costs.

One final observation I would like to offer is that it is not necessary to use only the so-called 24-hr estimating method for evaluating H and t_m in Equation 8 to arrive at an estimate of operating hours. Any accurate method of calculating average heat gain at a given outdoor temperature should suffice. The need for a heat gain and temperature estimate is to establish the slope of the line for a given structure which defines average heat gain *vs.* outdoor temperature.

C. M. TOELAER, Bloomfield, N. J. (WRITTEN): The study reported in the paper is a very interesting one and the conclusion may be substantiated by further observations. The present study was limited to eleven houses, 2 of which were studied for only 2 weeks and 4 for less than 2 months.

Residence 8 had an operating cost of $40.42 higher than the method anticipated or 25.37 percent above that predicted.

In Residence 11 the customer would have less reason to complain because the power consumption was 11.9 percent less than predicted. Another interesting observation is that Residences 9 and 10 in the same city were 12 percent high and 8 percent low respectively. When one examines the extreme deviations and observes a 37 percent difference from predicted between the high and the low, the conclusion might be drawn that the samples were too limited to attempt to substantiate the procedure.

It might be possible that consideration of other factors such as:

(1) percent sunshine hours; (2) wet bulb temperatures and their effect on comfort temperature; (3) the desired indoor temperature; and (4) the sensible capacity of the equipment with respect to load, would give more accurate results.

A method given by T. N. Willcox* gives estimated operating hour figures for various cities based on these other factors which seems to have a major bearing on operating cost.

Another factor that also has a bearing on operating hours are the user living habits and their desires as to temperatures and the use of outdoor air for night cooling. The method used by the authors include only one of the factors—outdoor temperature and uses the average daily temperature rather than the number of hours when the temperature is above a predetermined point. I believe we all accept the fact that lower indoor temperatures in summer mean higher operating cost just as higher temperatures in winter increase operating costs. An analysis of United States Weather Bureau data for any city will show considerably more hours above 75 F than above 80 F. This is not a straight line function as summer weather can be plotted on a probability curve with good results. This factor has also been neglected by the method.

The use of a 70 F datum temperature on the basis that a house remains at 5 F above outdoor temperature over a 24 hour period without air conditioning does not seem to me to be justified because during the hours when air conditioning is required the house (without air conditioning) is normaliy at or below outdoor temperature.

It is also noted that owners were asked to keep their thermostats between 72 and 75 F (average 73.5). I question the assumption and use of such a low indoor temperature

* Annual Utility Cost Operating Residential Cooling System, by T. N. Willcox (*Fuel Oil News*, Vol. 19, June 1954).

(and a fixed temperature as a basis) because the user preference survey reported in the previous paper indicated a general desire for higher temperatures.

The use of such a low temperature could make for reasonably consistent though not conclusive results because the lower the indoor temperature on which the load is calculated, the less the percentage effect of internal loads and solar load. A basic error in the method I believe is that it assumes loads which are independent of temperature difference are a function of temperature difference.

Although there may be some basis for using a 24-hr integrated load for the estimating of operating cost, it would seem to me that a separate load calculation for this purpose represents unnecessary additional work for that contractor. The industry has accepted sizing procedures which consider the 4- to 6-hr average peak load. The paper just presented on analogue computer analysis further substantiates this procedure.

The industry and the United States Weather Bureau must work together to arrive at suitable factors to be applied for computing operating cost based on the established load procedures.

AUTHORS' CLOSURE (L. A. Hall): Mr. Gilkey agrees in his comments that the datum temperature would be lower than the controlled room temperature. The paper states that when the average outdoor dry bulb equals the datum temperature, the net load over a 24-hr period will be zero. Since the solar and internal loads represent heat gains, the infiltration and transmission load components must be heat losses to obtain a heat balance.

The discussors kindly gave us advance notice of their finding of a 65 F datum for the test residence in comparison with our finding of 71.1 F. Rechecking our calculations we were unable to find any discrepancies in the original results. Specific details of the preliminary analysis method they used would be very desirable.

We heartily agree that there is a need for a realistic method of calculating residential cooling loads for THE GUIDE.

With reference to Mr. Chewning's comments, which the authors appreciate, we wish to emphasize that the datum found applies to residences only. As yet we have not attempted to apply this method to commercial buildings. For such buildings it can be expected that the datum temperature will be below 70 F.

In reply to Mr. Diehl, the paper is not at all restricted to a single compressor type of unit. The average capacity C and power input P over the season are values which must be obtained from the equipment manufacturer. If the two compressor approach will result in lower operating cost, and I am skeptical that it will, the saving will show up in the value of P used in the equation. The fact that only half of the capacity is used when needed does not necessarily result in a reduced operating cost. Over a 24-hr period a certain quantity of heat must be removed. One cannot state that two compressor operation will extract this quantity of heat at a lower operating cost than a single compressor unless information is known about the relative efficiencies. Moreover, as far as less frequent cycling of the two compressor unit is concerned, the thermostat differential is the preponderant influence on how frequently a unit cycles.

Mr. Ellenberger inquired whether an attempt had been made to evaluate the predicted power with datum temperatures other than 70 F. Datum temperatures from 65 to 75 F were investigated but it was found that 70 F resulted in the best correlation of the test data. If the datum temperature is taken as 75, the denominator of Equation 8 will be reduced but the numerator will be reduced more with a net effect of predicting less than the actual number of operating hours.

Mr. Ellenberger suggests that more weight be placed on lower values of $TC/24H$ in Fig. 2. The objective is to determine the datum which will correlate data collected over long periods. Since the higher values in Fig. 2 represent long hours of operation, their influence on the data should be greater than the very low values which

contribute very little to the total operating hours during a season. Therefore, if I were asked to weight these points, I would give greater weight to the higher rather than the lower values.

Since the Illinois people suggest the datum temperature should be 65 F and Mr. Ellenberger suggests 75 F, our value of 70 looks very good indeed.

An attempt was made to select test residences near local airports to enable us to use Weather Bureau data. The two residences tested in 1952 were completely instrumented to supply a continuous minute-by-minute record of outdoor weather conditions which were used.

As Mr. Ellenberger points out it is not necessary to use the so-called *24 Hr Estimating Method* to evaluate H which appears in Equation 8. This is also pointed out in the paper. The value of H merely represents the average hourly cooling load over a 24-hr period. However, it is important to note that use of a maximum instantaneous heat gain value for H in this equation will predict too large a number of operating hours.

The discrepancy concerning Residence 8 questioned by Mr. toeLaer has been explained in the paper. Also, as stated in the paper, no trend due to percent sunshine was found during the analysis.

As to the effect of wet bulb temperature, it is pretty well agreed that a variation between 40 and 60 percent relative humidity is scarcely perceptible to people. Hence, comfort can ordinarily be described in terms of indoor dry-bulb temperature, and this can influence the operating cost. The effect of different indoor conditions is not accounted for, but it is also not accounted for in predictions of heating costs by the degree-day method. The purpose here is to develop a method that will predict with reasonable accuracy the operating cost during normal summer weather and under usual operating conditions. Adjustments will then have to be made for any special conditions anticipated.

Mr. toeLaer states that Weather Bureau data will show considerably more hours above 75 F than above 80 F with which we heartily agree. However, we do not agree that this factor has been neglected in the method because the data have been correlated with a single datum temperature.

A survey made by *A.G.A.* of some 325 people living in air conditioned homes as to their preference of thermostat setting during the summer showed the following results: Approximately 50 percent had thermostat settings at 75 or lower, with the greatest percentage desiring settings of 74, 75 or 76 degrees. Therefore, the setting used during this investigation does not seem out of line.

Mr. toeLaer is correct that internal loads do not depend on temperature difference. This fact is pointed out in the paper during the discussion of the possible variation of datum temperature due to the relative proportion of the internal loads as related to the exterior loads. It is subsequently concluded that the datum temperature can be considered constant with sufficient accuracy for all practical purposes.

1521

No. 1521

SPECIAL MEETING

NOVEMBER 22, 1954, NEW YORK, N. Y.

Pres. L. N. Hunter called the special meeting of the Society to order at 2:00 p.m. in the Washington Room of the Hotel Statler, New York, N. Y. There were ten members present.

He advised that the meeting had been called to count the votes on the proposal to change the name of the Society from THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

Notice of the proposed change was sent to all members together with proxy ballots for those members eligible to vote.

On motion of P. B. Gordon, seconded by A. J. Offner, IT WAS VOTED:

THAT the name of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS be changed to AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., and that the officers be, and hereby are, authorized and directed to file an appropriate certification of change of name.

The four proxies, L. N. Hunter, W. L. Fleisher, A. J. Offner and P. B. Gordon, cast 3,278 votes in favor, 654 against the motion. Two members present voted in person.

The report of the tellers, W. J. Olvany, Jr., C. H. Hiers, W. M. Heebner and J. C. Fitts, *chairman,* was presented by Mr. Fitts.

In favor of the name change, 3,280; opposed, 654; ineligible, 76; illegible signatures, 69; unsigned proxies, 2. President Hunter stated that the eligible voting membership of the Society numbered 5,919 and 2,961 votes were required for a majority. As the affirmative vote was in excess of this number, namely 3,280, he declared the motion passed.

The meeting was adjourned at 2:20 p.m.

In Memoriam 1954

Name	Joined
Lawrence Lee Anthes *(Life Member)*, Toronto, Ont., Canada	1935
Joseph A. Archambault,* Sherbrooke, Que., Canada	1949
George F. Ball, Des Moines, Ia.	1951
William E. Baxter, Montreal, Que., Canada	1939
Thomas H. Bell, Columbus, Ohio	1944
Hermann S. Bogaty *(Life Member)*, Philadelphia, Pa.	1921
Clifford A. Booth, Toronto, Ont., Canada	1942
William W. Bradfield, Grand Rapids, Mich.	1926
Jess L. Brainerd, Seattle, Wash.	1948
Robert L. Bridgman, Silver Spring, Md.	1945
Robert Bruen, Oakland, Calif.	1945
Charles R. Campion, New York, N. Y.	1944
Delbert V. Case, Atchison, Kans.	1937
Clyde T. Clapperton, Toronto, Ont., Canada	1944
Robert H. Clark, Huntington Park, Calif.	1950
L. Villere Cressy, New Orleans, La.	1940
Mark C. Dreyer, Aurora, Ill.	1946
William H. Feirn, Madison, Wis.	1938
H. Allen Froelich, Junction City, Kans.	1939
Ray L. Gibboney, Toledo, Ohio	1954
George F. Goblaskas, Worcester, Mass.	1954
James P. Griffin, Tampa, Fla.	1951
E. Holt Gurney *(Presidential Member)*, Toronto, Ont., Canada	1929
Harold J. Hofmeister, Columbus, Ohio	1946
Richard E. Holmes, Springfield, Mass.	1949
Wilbur C. Kelly, Portland, Ore.	1935
Joseph V. Koubek, Cleveland, Ohio	1944
Bates E. Landes, Des Moines, Ia.	1938
John B. MacFerrin, Houston, Tex.	1944

* Died in 1953 but Society not notified until 1954.

In Memoriam 1954

(continued)

Name	Joined
William Mallis *(Life Member)*, Seattle, Wash.	1914
J. Harvey Manny, Chicago, Ill.	1936
William D. Marshall, Arlington, Va.	1935
E. McLaughlin, Phoenix, Ariz.	1953
Linn W. Millis *(Life Member)*, Kansas City, Mo.	1918
Fred G. Olson, Toledo, Ohio	1954
Oran W. Ott, San Marino, Calif.	1925
William S. Peters, San Jose, Calif.	1944
Fred W. Powers *(Life Member)*, Chicago, Ill.	1911
Henry P. Reger *(Life Member)*, Chicago, Ill.	1934
Charles F. Roth, New York, N. Y.	1930
Park E. Sleister, Omaha, Nebr.	1943
Clarence T. Sluder, Washington, D. C.	1944
J. Howard Smart, New Britain, Conn.	1944
Jay W. Snyder *(Life Member)*, Detroit, Mich.	1917
Alvin D. Stokes, Baltimore, Md.	1936
Thomas N. Thomson *(Life Member)*, Huntington, L. I., N. Y.	1927
Homer J. Tilley, Los Angeles, Calif.	1950
Bernard E. Tiltz, New York, N. Y.	1930
Reuben N. Trane *(Life Member)*, La Crosse, Wis.	1915
Silas M. Trumbo, Chicago, Ill.	1926
Walter G. W. Turno *(Life Member)*, Newark, N. J.	1912
J. Rexford Vernon, Milwaukee, Wis.	1926
Eugene Vivarttas *(Life Member)*, New York, N. Y.	1910
Joseph H. Volk, Milwaukee, Wis.	1923
Albert E. Vroome, Shreveport, La.	1932
Allen W. Williams *(Life Member)*, Columbus, Ohio	1915
Calvert C. Wright, University Park, Pa.	1944
Henry A. Zanone, Louisville, Ky.	1943

E. Holt Gurney

1882-1954



Members of the Society were saddened to learn of the passing on Friday, March 19, 1954, of one of its distinguished past presidents, E. Holt Gurney, an engineer and industrialist of Canada.

Mr. Gurney, the first Canadian to become president of A.S.H.V.E., joined the Society in 1929, and was an intensely interested member. He was active on Council from 1931 to 1935, and again in 1939. He was known to all members of the Society for his vigorous service on numerous important committees. From 1932 to 1934, he was a member of the Executive Committee, in 1933, the Membership Committee, and in 1935 was chairman of the Finance Committee and in 1936 a member of that committee.

Mr. Gurney was second vice president in 1936, first vice president a year later, and in 1938, became the Society's president. His administration was an aggressive one, marked by expansion of Society endeavors and increase in membership. When his term of office was ended he did not relinquish his enthusiasm, and for 1939, took over the chairmanship of the Advisory Council.

He was instrumental in the adoption of resolutions to improve procedures for election of officers and was on the Committee to Study Methods of Selecting Society Officers and Council Members. From 1937 to 1941 he was a member of the F. Paul Anderson Award Committee, and in 1942, an earnest worker on the War Service Committee. Three years later, he actively participated as a member of the Committee on Research Fund Raising. He never refused to undertake a task that would benefit the Society, and he, accompanied by Mrs. Gurney, was faithful in attending Annual and Semi-Annual Meetings.

Mr. Gurney was born in Toronto, Ont., Canada, in 1882, and received his elementary and preparatory education in that city. He was a graduate of the University of Toronto. During World War I he was director of the Imperial Munitions Board. He was vice chairman of the Ontario Research Foundation, a member of the Council of the *Royal Canadian Institute* and the *Institute of International Affairs*. He was associate director of the Canadian National

Exposition, and a former director of the *Canadian Institute of Plumbing and Heating.*

The University of Western Ontario, London, Ontario, doubly honored Mr. Gurney on June 4, 1949. He was selected to make the convocation speech before the graduating class of 900 students, and was given an honorary LL.D. Presentation of the degree was made by Arthur Ford, Chancellor of the University.

Funeral services were held Monday afternoon, March 22, at 2:00 p.m. and interment was in Toronto. He is survived by his wife, Nora Sutherland Gurney, and three daughters, Mrs. Peter Smellie of Ottawa; Mrs. Carr Hatch and Miss Betty Gurney of Toronto.

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